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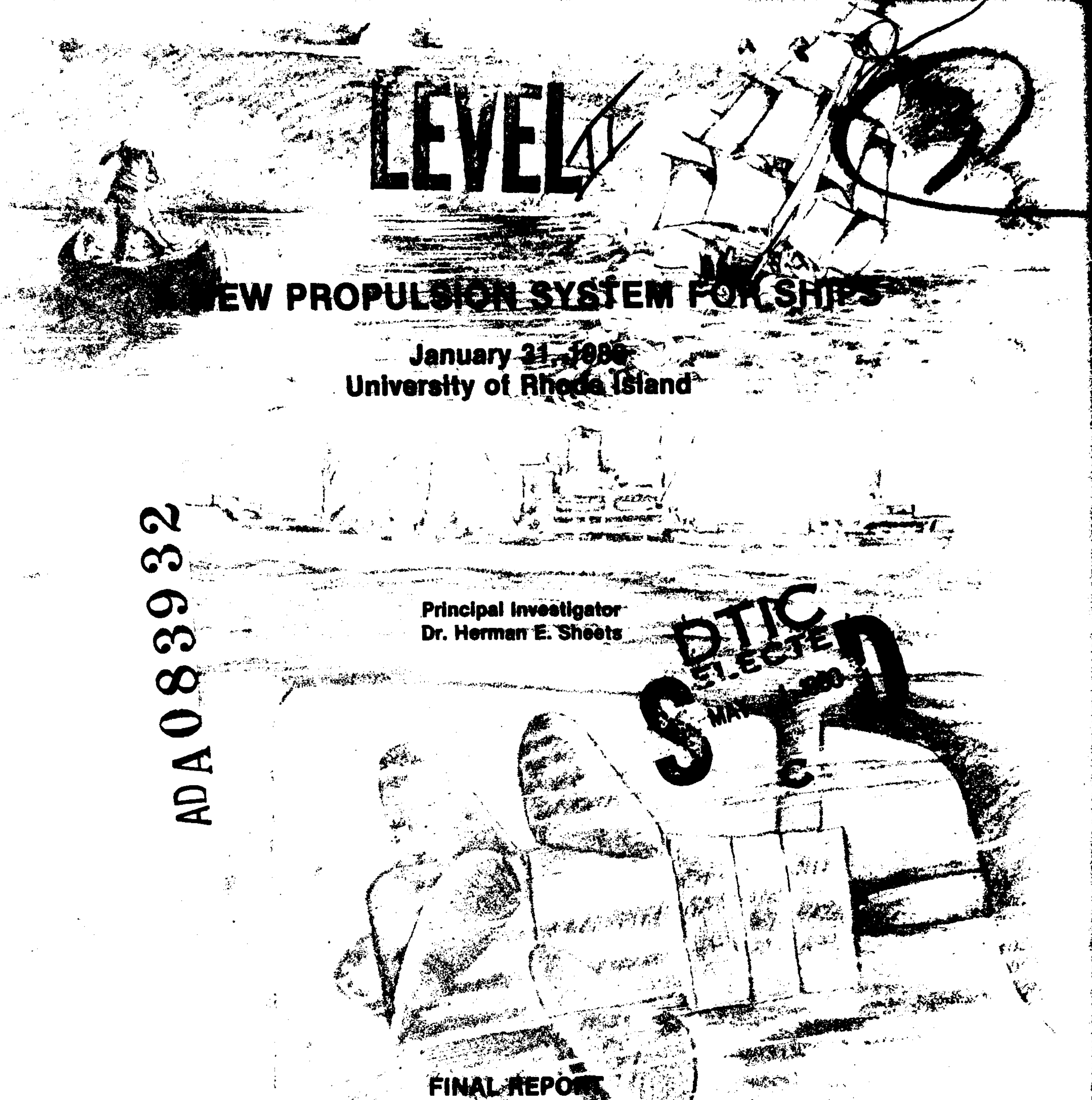

A NEW PROPULSION SYSTEM FOR SHIPS

January 31, 1980
University of Rhode Island

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Principal Investigator
Dr. Herman E. Sheets

FINAL REPORT

United States Navy — Office of Naval Research — Code 230
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NOMENCLATURE

v_m	ft/sec	Axial velocity
u	ft/sec	Circumferential velocity
w	ft/sec	Relative velocity
A	ft^2	Pump flow area
D	inch	Diameter
H	ft	Pump head delivered
Q	ft^3/sec	Pump flow capacity
n_s	-----	Specific speed
N_s	$gal^{½}/min^{3/2}ft^{3/4}$	Specific speed
n_{ss}	$gal^{½}/min^{3/2}ft^{3/4}$	Suction specific speed
ΔP	lbs/ft^2	Pressure increase
α	degree	Blade angle
β	degree	Blade angle
ρ	$lbs\ sec^2/ft^4$	Water density
Φ	-----	Flow coefficient
Ψ	-----	Pressure coefficient
η	-----	Efficiency
n	1/sec	Pump rotational speed
N	HP	Pump horsepower
H_{sv}	ft	Suction head at pump inlet

I. EXECUTIVE SUMMARY

This report covers the final phase of research on an entirely new concept for ship propulsion. In the slightly over five years since its inauguration, the University of Rhode Island has sought to develop a better understanding of the complex relationships involving ship propulsion, ship control and a host of independent problems related to hydrodynamics, structural mechanics, efficiency, weight and cost.

Shortly after the initiation of the program, it became apparent that the new concept offered a great potential for advancing ship propulsion to a new plateau. The pages that follow in the body of this report contain the more detailed results of the research that are summarized below.

The new propulsion system uses a hydraulic transmission, located outside the ship's hull, driving the propeller. Thus, the reduction gear and its foundations are eliminated. The transmission uses seawater as the motive fluid, and the fluid discharge from the turbine contributes to the thrust of the propulsion system. The propulsion system consists of a prime mover (inboard) coupled directly to an (outboard) axial flow pump. Pressurized water is produced by the pump to drive a multistage turbine that is located in the hub of the propeller

and forms an integral part of the propeller. The entire weight of the transmission turbine and propeller is on the same order of magnitude as the existing propeller for the same horsepower. There is a hydraulic thrust reversing mechanism incorporated in the transmission. The basic concept is shown in Figure A.

Since the concept is entirely new, a proof of concept air model was built and successfully operated (with a 39:1 speed reduction ratio) to demonstrate the structural principle of this system. A second air model was built, shortly thereafter and successfully demonstrated the feasibility and simplicity of contra rotation propellers with this propulsion system.

With proof of concept established, a water model of 10 horsepower was built and operated to demonstrate the entire system and get experimental data on efficiency. The single stage pump showed efficiencies between 85% and 90%. The five stage turbine of the water model showed an efficiency of 91%. The entire transmission showed efficiencies of 71% with speed reductions on the order of 6:1 and up to 8:1.

On the basis of the test results and the experience gained during the engineering phase, a transmission system with 80% efficiency can be attained in the 10 horsepower size with a four inch pump and a 5.7 inch turbine. The experimental data clearly indicated that a large transmission can be designed and built with efficiencies above 90%.

By the end of the third year, preliminary conclusions had been reached that this concept so far is more promising than initially anticipated. That feeling was reinforced by outside interest in the work, by presentations given and papers offered at scientific and development conferences where people from different backgrounds and disciplines were brought together. The work thus far did raise some questions about cost and efficiency that were examined.

An analysis was made to compare the "New Propulsion System for Ships" with the standard gear transmission and the results indicate that the new propulsion system equals and with design improvements it can exceed the efficiency of the standard gear transmission. Further improvement of the new propulsion system over the standard gear transmission is the result of an increase in thrust due to the transmission turbine discharge flow. It became evident that the new propulsion system can generate a higher thrust for the same propeller diameter, and it can produce a higher propeller efficiency for the same thrust. The new propulsion system has weight savings on the order of 1.5 times the reduction gear weight and there are equivalent savings in space. These savings can be used to improve ship's performance, increase pay load, or a combination of these factors.

A cost analysis has shown that the new propulsion system will be less expensive than a gear transmission. This is easily

understood due to the substantial reduction in weight of the new propulsion system compared to the gear transmission. The boundary layer control effects of the new propulsion system on the ship's hull and the propeller have been assessed but not quantified, and they provide additional improvements in overall ship system efficiency.

And finally, at the end of the five years the new ship propulsion concept has been studied, analyzed, and a research model has been built and tested. The studies have indicated that the optimum performance of the new concept should include the entire ship system, namely ship configuration and ship control in addition to ship propulsion. The transmission pump can be used for boundary layer control on the ship's hull and lead to entirely new and more efficient hull configurations. The transmission turbine discharge flow can be vectored in the desired direction, thus eliminating the ship's rudder. The concept is entirely new, and as with any new undertaking, additional improvements can be made with research on the reversing system, the transmissicn, as well as the integration of the propulsion system into the ship.

However, the data indicate it is now beyond any question or doubt that the concept is sound and offers a great potential improvement in ship design, propulsion and control. Like any new concept its full realization is incremental, moving from

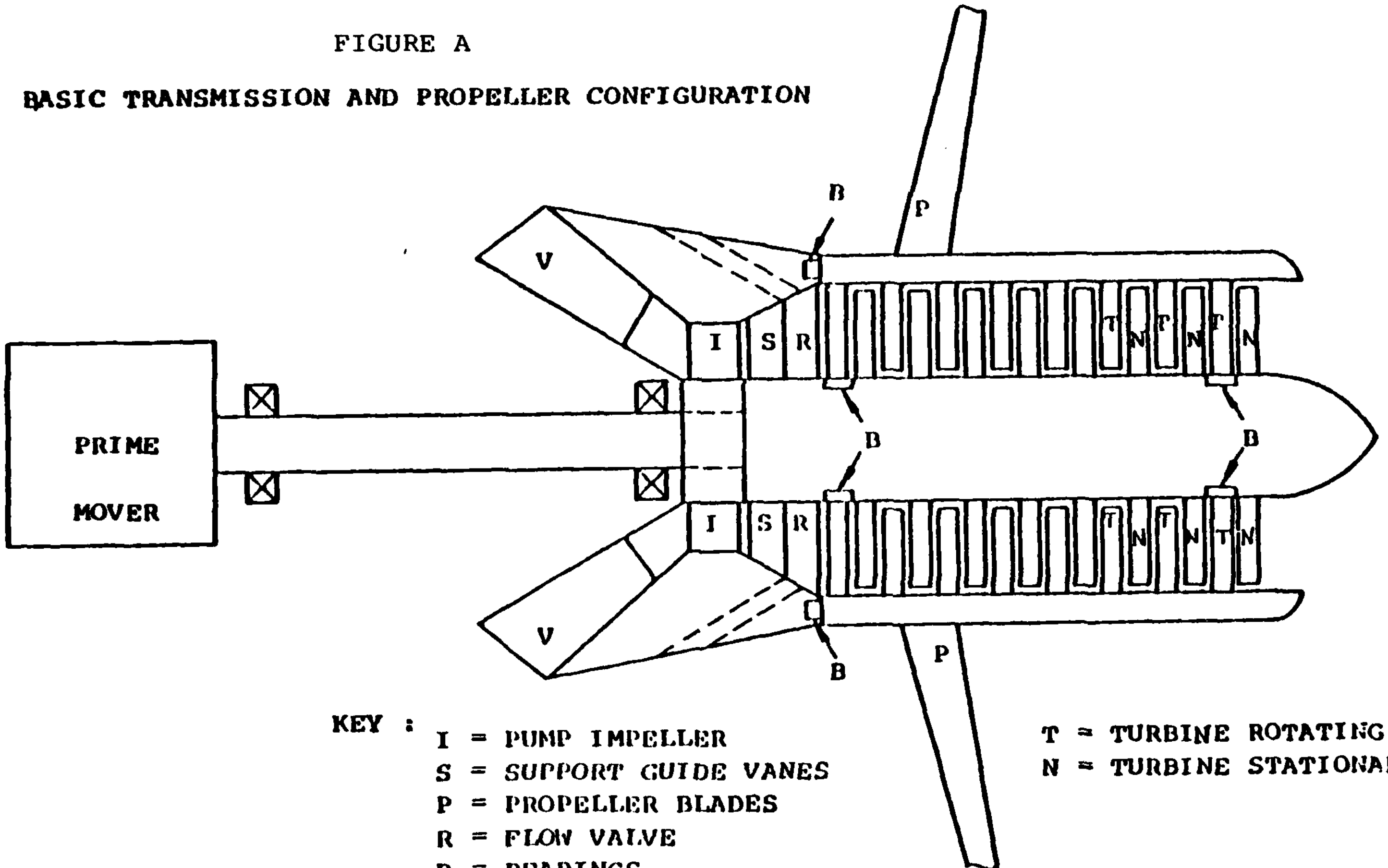
one phase to another in a logical sequence. A tentative conclusion to be drawn from the work done thus far is that it is entirely possible to advance ship design and propulsion to a new level of performance through the implementation of a three point program extending over three or four years costing approximately one million dollars to 1) design and 2) build a scaled up model that will be 3) tested at sea. There is an abundance of research, analytical and experimental data to substantiate the validity of this recommendation as being the next logical step to take.

The major features of the new propulsion system for ships are shown in the following figures. Figure A shows the basic transmission and propeller configuration including the reversing system. Figure B shows the axial flow transmission installed in a ship with contra-rotation propellers. Figure C is a picture of the contra-rotating air model of the new propulsion system. This model has been operated and tested. Figure D represents the concept of ship control by means of the new propulsion system, thus replacing the conventional rudder.

This project has been supported by and has been under the broad direction of the Office of Naval Research (Code 230) and the Maritime Administration, Marine Science Program. The project leader since its inception is Dr. H.E. Sheets, Professor of Ocean Engineering, with the assistance of several of the University of Rhode Island faculty and graduate students. Our special thanks goes to Mr. A. Davis for his extraordinary contributions in building the models and supervising the tests.

FIGURE A

BASIC TRANSMISSION AND PROPELLER CONFIGURATION



KEY :
I = PUMP IMPELLER
S = SUPPORT GUIDE VANES
P = PROPELLER BLADES
R = FLOW VALVE
B = BEARINGS
V = INLET VANES

T = TURBINE ROTATING VANES
N = TURBINE STATIONARY VANES

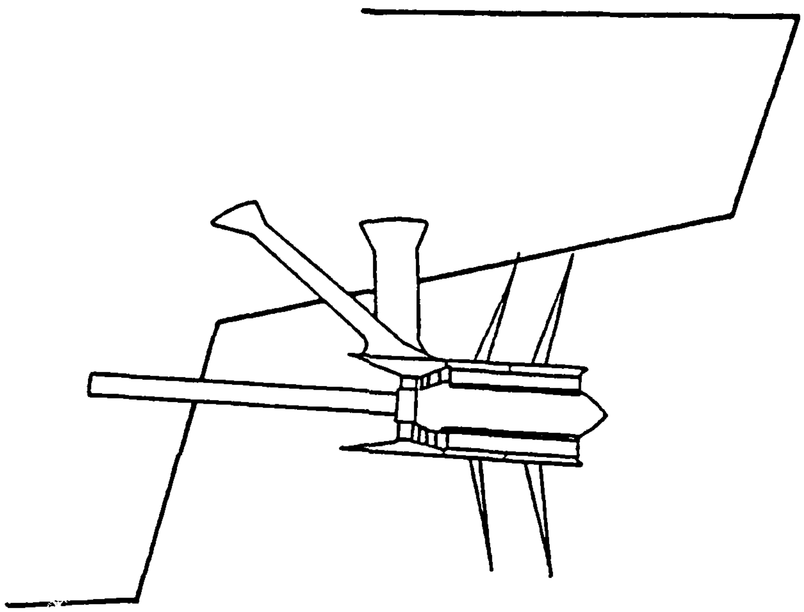


FIGURE B

The Axial Flow Transmission Used for
Counter-Rotation Propeller Propulsion

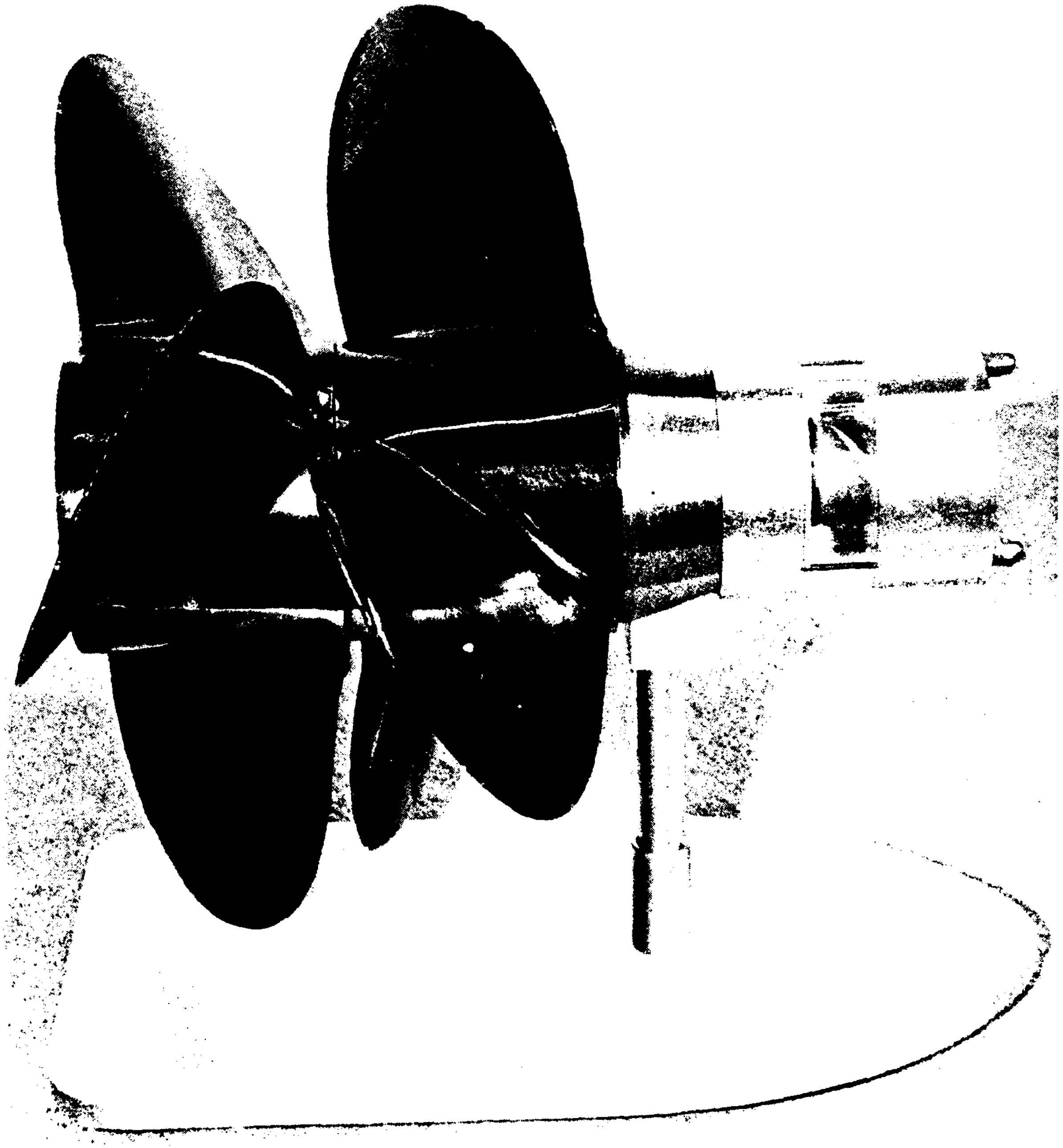


FIGURE C. Contra Rotating Air Model

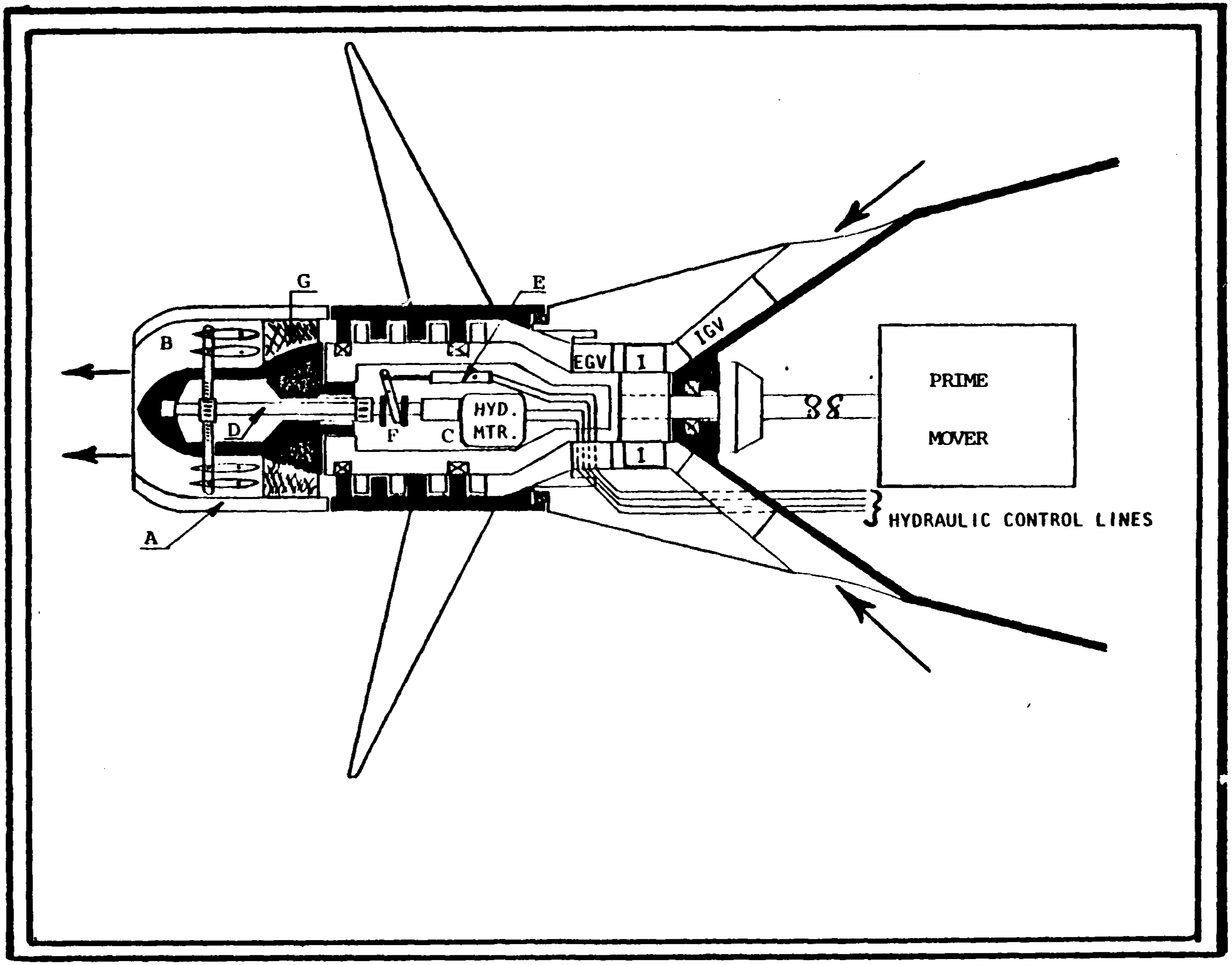


FIGURE D.   Ship Controlling Propulsion System

II. INTRODUCTION

A. Limitation of Existing Propulsion Systems

Propulsion systems for ships have been developed over a considerable period of time and have reached a status of great reliability. However, their weight and space requirements have become very large, particularly with increasing power requirements. As a result, a number of studies have been made to analyze and possibly overcome the limitation and performance shortcomings of existing ship propulsion systems. Light weight propulsion systems for naval ship applications have recently been discussed by Kuo (1977, 1978, 1979). These reports concentrate on open and closed cycle gas turbine power plants associated with a number of propulsion and speed conversion systems to transmit the power from the prime mover into the water. The reports present a survey of power requirements of a number of ships. They cover both multiple shaft and single shaft ship propulsion systems. Thruster performance and associated weight estimates are given for subcavitating propellers, supercavitating propellers, water jet propulsion together with appropriate comparison diagrams. This report also contains detailed information on gear weight and volume as well as propulsion shaft weight. It lists performance estimates together with component size, weight,

and costs for a range of propulsion power requirements. Separately, system alternates and critical technologies are discussed. A different analysis of ship propulsion systems is discussed by Hauschildt (1977). This report analyzes alternate possibilities of ship propulsion for the DD-963 destroyer, and it also reviews propulsion system requirements for surface effect ships, hydrofoils, and other high performance vessels. The report covers special gears, advanced electrical transmission systems, contra-rotating propellers, and super conductive electric drives. Both Kuo and Hauschildt indicate that the basic gear transmission driving a slow speed propeller is essentially a propulsion system which has not changed for many years and still is an efficient propulsion system for the ship speeds which are economic and in use in the majority of commercial and Navy craft. Whereas new technology has been introduced in many other ship systems, including the prime mover, the propulsion system has had very little change. Kuo (1978) has studied the preliminary design of an 80,000 horsepower closed cycle helium gas turbine driving a propeller through a reduction gear. While this prime mover has advantages over existing prime movers, the system uses a reduction gear requiring substantial weight and space. Kuo (1979) discusses electrical propulsion systems, including superconducting electric transmission systems and the critical technologies for ship propulsion. These studies address

themselves essentially to improvements of the prime mover but show no innovation regarding the power transmission system. The propulsion system using reduction gears is heavy and requires a substantial amount of space. In considering alternate propulsion systems, high efficiency and reliability are mandatory.

B. <u>Trends in Ship Power Requirements</u>

During the last decade, the size of various ships has increased, and with ship size, the power requirements have also increased. For naval applications, an increase in ship speed is desirable, and this requires an increase in power. In addition, there has been a tendency for single screw propulsion wherever feasible and advantageous. Single screw propulsion has been discussed by Hydronautics (1979), and this report expects a growth in power for single screw ships. It is noted in this report that higher powered single screw ships can have vibration problems as well as compatibility problems between hull and machinery. Recently, the increase in fuel costs has resulted in a number of discussions and papers (Lewis, 1977). These papers clearly indicate that the increase in fuel costs require substantial attention to more efficient power plants and propulsion systems. While a number of suggestions have been made to increase the efficiency of prime movers, the suggestions to the propulsion systems have been very limited. It is well known that a reduc-

tion in propeller speed will give improved propeller efficiency at the expense of increased weight, volume, and costs of the gear transmission. Therefore, the selection of the shaft speed reduction system is usually a careful compromise in efficiency, weight, and cost.

C. Propeller Configuration, Supercavitating Propellers, Water Jet Propulsion

Subcavitating propellers are still the main propulsion system to convert shaft horsepower into thrust because they give the best efficiency and lowest cost for the most commonly used ship speeds and sizes. Figure 1 indicates propeller efficiency as a function of rotative coefficient. It also indicates the range of efficiencies being on the order of 70% to 76% as the maximum. In order to reduce local thrust variations and reduce the effects of structure supports on the propeller, propellers have been designed having skewed blade arrangements. While these configurations are more difficult to manufacture and have some limitations in terms of stress and structural rigidity, they have become quite popular in recent years for single screw ships. However, it is noted that for high efficiencies, propeller diameters become large, and propeller shaft speed becomes increasingly slow, particularly for ships of moderate speed. This is

indicated in Figures 2 and 3 showing propeller diameter in feet as a function of thrust power together with propeller shaft speed.

Supercavitating propellers have been discussed in the literature, and they certainly may find applications for high speed ships of 50 knots and above. Figure 4 shows propeller diameter and propeller shaft speed as a function of thrust power.

Water jet propulsion has been introduced for some small vessels, and it is generally agreed that the efficiency attained by water jets is affected by many more factors than propellers. The literature indicates that essentially water jets do not provide the same overall propulsive efficiency as supercavitating propellers. Their optimum application is for high speed ship propulsion on the order of 50 knots and above. However, the pump of the water jet may be able to operate at a higher speed than the propeller and, therefore, may result in less technical demands on the gear reduction, possibly resulting in a smaller gear of less weight and space. All propulsion systems must also consider a range of operation which covers off-design engine operating characteristics and ship operations in various sea states. Another variable is the increase in drag of the ship's hull due to fouling and other operating conditions. Under these conditions, the variable pitch propeller has distinct advantages.

D. Integrate Ship and Propeller Into a Single System

The above reports, Kuo (1977, 1978, 1979), Hauschildt (1977), Hydronautics (1979), and Lewis (1977) cite many references describing the present state of the art in ship propulsion. Generally, the various studies relating to ship propulsion have addressed themselves basically to the three areas -- prime mover, reduction of shaft speed, and propulsor. Only in the case of water jet propulsion, some consideration is being given to integrate the ship configuration and the propulsion system into a single system. Unfortunately, the efficiency of water jet propulsion is only competitive for very high speeds and, therefore, the system has been applied only to surface effect ships and some very small specialty vessels. However, with the increased attention for achieving maximum overall efficiency and low fuel consumption, it will be beneficial to analyze a ship propulsion system which can address itself to integrate the ship and the propeller into a single system for optimum efficiency. The proposed new propulsion system for ships is an effort to integrate ship, propeller, propulsion, and power plant into a single system. The new propulsion system proposes an axial flow hydraulic transmission located outside the ship's hull driving a propeller. The transmission turbine is located in the hub of the propeller, and turbine discharge will contribute to the thrust of the propulsion system.

The study of the axial flow hydraulic transmission and the new propulsion system is described in section 3. In order to provide a proof of concept, two air models have been operated and are described in section 4. In addition, a water model has been built and tested. The water model is discussed in section 5. The test data on both the air and water models have been most encouraging regarding efficiency and performance of the new propulsion system.

III. THE NEW PROPULSION SYSTEM

A. The Hydraulic Axial Flow Transmission

The need for reduction of weight and space aboard ship is well known. In the search for such a reduction, various systems were studied, and a hydraulic axial flow transmission was considered for detailed analysis as it eliminates the requirement for a reduction gear. The basic concept of this transmission was first mentioned and studied by Sheets (1974, 1975). The new propulsion system uses a hydraulic transmission, located outside the ship's hull, driving the propeller. Thus, the reduction gear and its foundations are eliminated. The transmission uses seawater as the motive fluid, and the fluid discharge from the turbine contributes to the thrust of the propulsion system. The propulsion system consists of a prime mover, such as a steam turbine, gas turbine, diesel engine, or electric motor supplying the power. An axial flow pump is directly driven by the prime mover, preferably at the shaft speed of the prime mover and is using seawater as its fluid. An axial flow water turbine is driven by the pressurized water produced by the pump. A conventional propeller is driven by the water turbine. There are radial and thrust bearings for the support of the propeller. The pump has supporting vanes, a shroud, and a thrust reversing mechanism. Figure 5 shows the basic arrangement of the system.

Because seawater is used, there is no need for transmission cooling and heat transfer equipment. Figure 5 shows that the multistage axial flow turbine is located in the hub of the propeller and forms an integral part of the propeller. The rotating blades of the turbine are connected at their periphery to a rotating member which can be positioned and fixed inside the propeller hub. As a result, the stationary nozzles are supported from a stationary member around the center of the axis. Thus, the entire weight of the transmission turbine and propeller is on the same order of magnitude as the existing propeller for the same horsepower.

In marine propulsion systems, the required shaft speed reduction between prime mover and propeller may vary. With a turbine as a main power plant and a propeller as a propulsor, a speed reduction of about 20:1 is a reasonable value, and it may vary in the range of 10:1 to 40:1. A detailed description of the hydraulic axial flow transmission is presented in Sheets (1975, 1978-1).

B. Pump -- Specific Speed and Efficiency

The pump is part of the hydraulic transmission, and it will drive the hydraulic turbine. In analyzing the hydraulic transmission, a number of pump and turbine combinations have been studied, resulting in pump performance covering a range of pres-

sures, flows, and specific speeds. In these studies, both conventional pumps with an unsymmetric vector diagram as well as low cavitation pumps with a symmetric vector diagram have been considered, Sheets (1975, 1978-1).

For the hydraulic transmission, the efficiency is most important so that the transmission will be competitive with or exceed in efficiency existing systems. The efficiency of a hydraulic transmission depends on the individual efficiency of pump and turbine. However, the efficiency of pumps and turbines decreases with an increase in specific speed. While centrifugal hydraulic transmissions have been built with total efficiencies of approximately 90%, such efficiencies must be demonstrated by tests for axial flow transmissions. If the transmission ratio of the axial flow transmission is relatively high, then the specific speed of the axial flow pump is high and the specific speed of the turbine is low under otherwise similar conditions. Thus, the pump efficiency is the critical element for the total transmission efficiency. Published data in textbooks indicate centrifugal pump efficiencies of 90% and slightly above in a certain range of specific speeds and for large values of flows, but they show a significant reduction in efficiency with increasing specific speed, as shown in Figure 6. However, considerable research has been done and advances in technology have been made on

axial flow compressors, and the use of laminar flow airfoil blades has been introduced resulting in increases in efficiencies and also permitting higher pressure increases per stage. There are no literature data available indicating the effects of laminar flow blades on pump performance. However, it can be expected that it will result in increases in efficiency. This increase in efficiency has been estimated and is shown as a dotted line in Figure 6 for pumps of large capacity and high Reynolds numbers. Thus, careful consideration must be given to specific speed, flow quantity, Reynolds number, and other factors in the selection of the pump in order to achieve the desired high efficiency.

Another consideration in the selection of the pump is the effects of cavitation. Cavitation can be evaluated using the concept of the suction specific speed. The equation for the suction specific speed, n_{ss}, Wislicenus (1965) is as follows:

$$n_{ss} = \frac{n\ Q^{1/2}}{H_{sv}^{\ 3/4}} \qquad \frac{gal^{1/2}}{(min^{3/2} \times ft^{3/4})} \tag{2-1}$$

Generally, carefully designed pumps of large size and flow quantity may operate cavitation free at a suction specific speed, n_{ss}, up to 15,000. Operation at higher suction specific speed values has been achieved by using inlet inducers or special inducer type pumps. They have been operated successfully up to suction specific speed value of 40,000, Sladky (1976). However,

flow quantity and Reynolds number also affect cavitation. A relatively large Reynolds number in the pump impeller will result in favorable cavitation conditions since cavitation tests have indicated that cavitation performance improves and allowable suction specific speed increases with increased pumped volume and Reynolds number, Anderson (1961). It can be expected that the very small pumps will operate at high efficiency without cavitation up to a suction specific speed of $n_{ss} = 10,000$.

Inducer pumps with improved cavitation characteristics are not planned for the hydraulic transmission at this time as their performance results in a moderate reduction of efficiency. However, a pump with 50% reaction and a symmetric vector diagram may be considered as it offers high efficiency and improved cavitation characteristics. For axial flow pumps, a very small amount of cavitation may not be harmful to the performance since ship propellers frequently operate with a small amount of cavitation.

If the hydraulic transmission is installed in a ship, consideration must be given in the pump's cavitation analysis to the forward speed of the ship and the water level above the pump impeller, as both will increase the value of the pressure H_{sv} and will decrease the value of the suction specific speed n_{ss}.

Cavitation analysis for the hydraulic transmission considering a 15,000 horsepower propulsion system is discussed in

Sheets (1978-1). This analysis reviews a low cavitation pump with symmetric vector diagram. For such a pump, Equation (2-1) must be revised in order to consider the lower relative velocities over the pump blades. The data indicate that the new propulsion system is particularly applicable to higher speed ships.

C. The Inverted Turbine

A novel part of the hydraulic transmission is the use of an inverted turbine in which the rotating blades are structurally connected to a cylinder forming the outside of the turbine. This cylinder then forms an integral part of the propeller and is located in the hub of the propeller. As a result, the turbine nozzles are supported from a stationary shaft located in the center of the turbine and the propeller. This stationary shaft carries the bearings which support both the rotating turbine member and the propeller. The weight of the propeller and turbine is transmitted into the bearing through the first and last rotating row of the turbine. The axial thrust is carried to a thrust bearing through the rotating turbine cylinder forming the outer diameter of the turbine. The entire turbine structure is novel since such a design has never been proposed before. However, such a design permits an integral structure of transmission turbine and propeller, and it will result in substantial weight saving for the propulsion system. It will also result in

increased efficiency since the turbine discharge will contribute to the thrust of the entire propulsion system.

The hydraulic turbine must match the low shaft speed of the propeller, and it determines the power transmission ratio. At first an impulse hydraulic turbine has been considered for the transmission with a velocity ratio of $u_{tm}/c_{t1} = 1/2$. It is desirable to keep the diameter of the turbine small, and various diameters were analyzed.

In much of the turbine literature, the optimum efficiency is related to the velocity ratio of u_{tm}/c_{t1}. In other literature, a pressure coefficient is used. The relationship between these two factors is as follows:

$$H_t = \Psi_{tm} \frac{u_{tm}^2}{2g} = \frac{c_{t1}^2}{2g} \tag{2-2}$$

$$\frac{u_{tm}}{c_{t1}} = \frac{(1)}{(\Psi_{tm})}^{1/2} \tag{2-3}$$

It is noted that for turbines, the circumferential velocity, u_{tm}, and the pressure coefficient, Ψ_{tm}, relate to the mean diameter. Subsequently, reaction type turbine blading was studied with a velocity ratio, u/c_{t1}, on the order of 0.7, corresponding to a pressure coefficient of two. The reaction tur-

bine requires about twice the number of stages for the same pressure drop and diameter. However, reaction turbines have higher efficiencies. In addition, the reaction turbine will divide the pressure drop between nozzle and rotor blades. With a 50% reaction turbine, the pressure drop is equally distributed between nozzle and rotor blades.

There is a need to optimize the entire hydrodynamic transmission for shaft speed, water flow, pressure drop, turbine diameter as well as minimum number of turbine stages and efficiency. Additional considerations for the hydraulic transmission consisting of an axial flow pump followed by an axial flow turbine, are the thrust forces and the turbine exit velocity.

In order to assess the total capability of the hydraulic transmission, it was felt that good efficiency was most important. Therefore, a reaction turbine was chosen. The reaction turbine has lower flow velocities in the blades and less flow deflection. Thus, it has a higher efficiency. The reaction turbine has a pressure drop in the rotating blades and accordingly, a thrust force is developed in the rotating stages. If the turbine is located in the hub of the propeller and the rotating blades of the turbine form an integral part of the propeller, then the turbine thrust forces will reduce the propeller thrust forces so that the forces on the thrust bearing are reduced accordingly. As a result, the design shown in Figure 5 was

developed. Most turbines are designed for low exit velocity as the energy in the exit velocity is considered an energy loss. However, in the ship propulsion application, it is desirable to have the turbine exit velocity at such a value that it will contribute thrust to the total propulsion system. Thus, the turbine exit velocity should be about equal to the propeller discharge velocity. Preferably, the turbine rotating blade exit should be followed by a diffuser, similar to applications in large water turbines, so that the exit velocity is distributed evenly over the turbine exit diameter. Accordingly, the local velocity at the blade exit in the turbine annulus is higher than the velocity at the diffuser exit. The reaction turbine is well suited to meet the above requirements.

The transmission incorporates the turbine as an integral part of the propeller and thus, the rotating part of the turbine is a ring covering the tip of the blade whereas the stationary members are carried from stationary shaft, coaxial with the centerline of the propeller, as shown in Figure 5. Another feature of the design consists in the first and last rotating stages of the turbine each carrying an inner ring which supports the bearing carrying the entire turbine and propeller structure. The other turbine stages are carried by individual rings, and each ring is connected into the rotating structure or the adjoining turbine ring. The stationary turbine nozzles are also

carried by individual rings which are located axially downstream of each other to provide for appropriate location inside the entire turbine configuration. When assembled, the entire turbine is contained in an axial cylinder which is mounted inside the propeller and forms an integral part with it.

D. Integrated Transmission Turbine and Propeller

The total thrust forces of the propulsion system are developed by two components -- the propeller and the flow from the transmission turbine discharge. The discharge from the hydraulic transmission and its associated thrust can be chosen by the designer to meet the needs of the propulsion system. The larger the component of thrust becomes from the transmission turbine discharge, the smaller is the remaining component of thrust from the propeller and accordingly, the decrease in the amount of power to be transmitted to the propeller. A great number of combinations of pumps and turbines have been analyzed. These studies have shown that the flow through the hydraulic transmission is a function of the speed reduction ratio, the turbine diameter, the pump diameter, the number of turbine stages, as well as selection of reaction turbine blades versus impulse turbine blades, cavitation parameter and forward speed of the ship.

A preliminary analysis for the optimum distribution of thrust between the propeller and turbine exhaust is made based

on the simple momentum theory. For this analysis, the average velocity of the jet has been calculated at the outlets of both the turbine and the propeller disk. The turbine discharge velocity, v_{te}, for this calculation is assumed to be equal to the propeller discharge velocity, v_{jpr}, but other velocity ratios are also possible.

$$v_{te} = v_{jpr} \qquad \text{or} \qquad v_{te} = y\, v_{jpr} \tag{2-4}$$

y is an arbitrary factor

It must be noted that the propeller discharge velocity, v_{jpr}, varies both with the above ratio and with the turbine diameter. Calculations have been made for transmission efficiencies, η_{tr}, of 85% and 90% and for propeller efficiencies, η_{pr} equal to 70% and 76%. Since the transmission efficiency is higher than the propeller efficiency due to the specific speed values of pump and turbine, the thrust from the turbine discharge is generated at a higher efficiency. For comparison, the gear driven propeller efficiency, $\eta_{pr}{}^{1}$, must include gear efficiency of 98% and shaft and thrust bearing losses of 1% resulting in a mechanical efficiency of $\eta_m = 0.97$. Gear driven propeller efficiency:

$$\eta_{pr}{}^{1} = \eta_m \cdot \eta_{pr} \tag{2-5}$$

$$\eta_{pr}^{1} = 0.97 \cdot \eta_{pr} \tag{2-6}$$

Figure 7 shows by a dashed line a typical variation of the propeller thrust as a function of the propeller radius fraction, R/R_{max}. Calculation methods are presented in Saunders (1957). Figure 7 also shows the average value of thrust for the conventional propeller. The solid line in Figure 7 shows an example of the possible average thrust and the thrust distribution of the new propulsion system. The common propeller has no thrust in approximately the inner 20% of the radius near the hub, and even at a radius ratio beyond that point, the thrust values are low. The new propulsion system has improved thrust values over the entire radius of the propeller, and the thrust value from the turbine discharge can have relatively high and constant values. At the design point, the turbine discharge thrust has no circumferential flow component, and it is generated at a higher efficiency than the propeller thrust.

The turbine discharge diameter can be expected to be larger than the hub diameter of regular propellers. It can be expected that the velocity of the turbine discharge will act as boundary layer control on the region of the propeller near its hub. Thus, higher local lift coefficients and thrust values appear possible. The flow through the hub of the propeller has never been attempted in this manner and deserves further study and experimentation.

Nevertheless, due to the flow through the hub, the propeller blade design can be altered and the entire propeller carries a smaller thrust load. For constant propeller diameter, this will result in improved propeller efficiency, and the average thrust of the new propulsion system will be considerably lower than the average thrust of the regular propeller. As an alternate, it can allow a decrease in the propeller diameter due to the increase in thrust from the turbine discharge or possibly a combination of some decrease in propeller diameter and some increase in efficiency.

On the basis of published propeller data, Sheets (1978-1), it is estimated that for the new propulsion system the propeller efficiency increases with the thrust ratio, T_{te}/T_{tot}, as follows:

$$\eta_{pr}' = (0.15\ T_{te}/T_{tot} + 1.0)\ \eta_{pr} \qquad (2\text{-}7)$$

With the above equation, the total efficiency of the new propulsion system can be calculated. With a propeller efficiency of η_{pr} = 70%, the efficiency of the comparable gear driven propeller equals η_{pr}^{1} = 67.9% and for the propeller efficiency of 76% η_{pr}^{2} = 73.7%, both are shown in Figure 8 as horizontal lines. Figure 8 presents total efficiency η of the new propulsion system as function of the thrust ratio, T_{te}/T_{tot}. The increase

in propeller efficiency, η_{pr}', as a function of T_{te}/T_{tot} is shown in Figure 8 by η_{pr}' for the 70% propeller efficiency and by η_{pr}'' for the 76% propeller efficiency. The total efficiency, η, increases rapidly with T_{te}/T_{tot} and equals η_{pr}^{1} at about 0.15 t_{te}/T_{tot} for a transmission efficiency of η_{tr} = 90%. It is noted that the calculations have been made for $v_{te} = v_{jpr}$, and this is not the optimum value for the maximum total efficiency. For the optimum value, the total efficiency η equals η_{pr}^{1} at a lower thrust ratio value. For a transmission ratio of η_{tr} = 85%, the total efficiency increases at the same rate with increasing thrust ratio but the value for η_{pr}^{1} is reached at a higher thrust ratio value. Figure 8 shows that the total propulsion efficiency, η, of the new propulsion system increases so rapidly with T_{te}/T_{tot} that for values of T_{te}/T_{tot} about 0.10 and above there is barely a difference in total efficiency with the gear driven propeller provided the transmission efficiency is near 90%. Figure 8 assumes the same shaft speed of the propeller for the new propulsion system and the gear driven propeller. If the new propulsion system drives the propeller at a lower shaft speed or drives contra-rotating propellers, then the new propulsion system will have a higher efficiency than a gear driven propeller.

E. The Thrust Reaction System

The propulsion system has some special requirements regarding the mechanical design. The thrust bearing for the propeller can

be located outside the hull of the ship. Presently, the entire thrust load of the propeller is taken up by a thrust bearing inside the hull. For the proposed propulsion system, the thrust loads are shown qualitatively on Figure 9. The rotating reaction turbine blades generate a thrust force opposite the propeller thrust force and the stationary turbine blades generate a thrust force opposite to the stationary pump guide vanes. For a turbine and pump with 50% reaction blading, the thrust bearing outside the hull carries about half of the total thrust, T. The thrust forces of the stationary pump and turbine blades about compensate each other and the rotating pump impeller requires a thrust bearing inside the hull of the ship carrying the other half of the total thrust.

For a contra-rotating propeller with about equal load in both propellers and a 50% reaction turbine, all thrust forces of the propellers and the turbines compensate each other, and with a 50% reaction pump, half of the thrust force is transmitted directly through the pump guide vanes and the other half of the thrust force through the pump impeller, requiring a thrust bearing inside the hull of the ship. For the single propeller shown in Figure 9, turbine blades with a high degree of reaction can substantially reduce the thrust load on the thrust bearing outside the ship's hull and a pump with a low degree of reaction in the rotating impeller can reduce the loads on the pump thrust

bearing. The decisions regarding pump and turbine degree of reaction have to be carefully analyzed considering efficiency, thrust forces, specific speed, and stress level on the blades.

A study of the bearings has been conducted for both the thrust bearing and the radial-journal bearing in the hub of the hydraulic transmission. These bearings are located outside the hull of the ship, and for these bearings, water lubrication is most desirable, and this study analyzes such a bearing system. Bearings located inside the ship are not analyzed as they are of conventional design.

With the hydraulic transmission, there is a need for a thrust bearing inside the hull of the ship to take care of the thrust developed by the pump. This is a conventional bearing. Figure 9 shows an example of possible locations of the required thrust bearings. The single thrust bearing, T_2, is located inside the ship's hull, and a thrust bearing, T_1, is located outside the ship's hull on the turbine periphery. There are also two journal bearings, T_3. The analysis has shown that the design of the transmission can vary the load which the thrust bearings must carry.

The bearings external to the ship's hull have distinct advantages if plastics are used as the bearing materials together with water lubrication. These new plastic materials are essentially free from corrosion, operate quietly, are available in

various molded shapes and are compatible to many other materials which minimizes or even eliminates lubrication. With lubrication, plastic bearing wear rates are negligible, thus increasing the life of the bearings.

F. Thrust Reversal

By removing the reduction gear and connecting the pumpjet directly to the prime mover, it is recognized that in reversing the direction of the pump, the propeller rotation will not be reversed because the guide vanes in the pump and the nozzles in the turbine do not permit effective reversal of turbine rotation. Reverse thrust can be obtained by reversible pitch propeller blades or possibly by including a reverse thrust water turbine stage in the propeller hub. However, for the new propulsion system, a rather simple reversing system can be designed by arranging closing vanes downstream of the pump and have special reversing thrust openings in the shroud of the pump. This arrangement provides a simple way to reverse thrust and is shown in principle in Figures 10 and 5. By closing the normal flow discharge path of the pump, its capacity will be reduced with a simultaneous increase in pressure, thus permitting an effective flow discharge through the reversing nozzles.

The hydraulic transmission permits the incorporation of such a hydraulic thrust reversing mechanism. This is accomplished by reversing the flow leaving the pump, and several designs are possible. The flow from the pump to the turbine is closed so that the turbine and propeller receive no motive force. The thrust in the reversed direction will not equal that in the forward direction but will be large enough to permit the maneuvering of the ship under a wide variety of conditions. It can be expected that such a hydraulic thrust reversal system can be activated much faster than a reversing turbine or a reversible pitch propeller.

G. Combined Propulsion Efficiency and Transmission Ratio

The propeller driven by the new hydraulic transmission can have a higher efficiency than a gear driven propeller in a certain operating range due to the water discharge from the turbine in the hub of the propeller giving additional thrust as mentioned earlier, Sheets (1978-2). Figure 7, Sheets (1978-2), shows that the new propulsion system results in a lower average thrust value than a conventional propeller of the same diameter, thus resulting in improved propeller efficiency for the thrust generated by the propeller. The total efficiency of the new propulsion system is presented in Figure 8. In the analysis of the performance of the new propulsion system, the assumption has been

made that the propeller diameter of the new propulsion system and the conventional propeller are identical. It has also been assumed that the average turbine discharge velocity going through the hub of the propeller is identical to the average propeller axial discharge velocity.

The total propulsive efficiency has been studied by analyzing the change in propeller efficiency as a function of the ratio of turbine discharge thrust to total thrust T_{te}/T_{tot} between the values 0 to 0.3 for three individual propellers of the Wageningen B4-40 and B3-35 series. Three propellers have been analyzed with efficiencies of 65%, 70%, and 76%, Sheets (1978-2). For each condition, the analysis was made by determining values from the Wageningen charts for a number of points within the above thrust ratio to analyze the increase in efficiency for a constant diameter propeller which will develop increasingly less thrust as the ratio T_{te}/T_{tot} increases. The data indicate that for practical purposes, there is a linear relationship between propeller efficiency and the function T_{te}/T_{tot}. The relative increase in efficiency is somewhat larger for the 65% efficient propeller as compared to the 76% efficient propeller. The above analysis justifies the use of Equation (2-7) for the range of propeller efficiencies and thrust ratio T_{te}/T_{tot}.

It is well known that propeller efficiency increases with propeller diameter and that increased propeller diameter requires a slower shaft speed. In many gear transmissions, it is

customary to have a two step reduction gear or otherwise limit the speed reduction to certain values so that the gear will not become too large and too expensive. For high values of speed reduction, such as 40:1 and larger, the gear can become excessively large, heavy, and costly. In the case of the new hydraulic transmission, there is considerably more liberty in selecting the speed step down ratio. A larger step down ratio may require more turbine stages and perhaps a larger turbine diameter, but these features are generally not very costly and will not increase the weight of the transmission by a substantial amount. An analysis has been made based on the Wageningen B4-40 and B3-35 propellers to analyze propeller efficiency as a function of diameter ratio and shaft speed. In this case, the basis for the analysis is a propeller of 70% efficiency. Table 1 shows the calculated data of the increase of efficiency as a function of propeller diameter ratio and changes in shaft speed ratio. Figure 11 presents the calculated data which indicate that it is very beneficial for the new propulsion system to drive a propeller of relatively large diameter and make use of the increase in propulsive efficiency permitted by such a design. The propulsive efficiency can be increased by over 10% if the propeller diameter is appropriately increased and the shaft speed reduced.

The previous analysis, Sheets (1978-1) was made by assuming that the average turbine discharge velocity, v_{te}, is identical

to the average propeller discharge velocity, v_{pr}. Additional calculations have been made varying this velocity ratio and also the ratio of the turbine diameter to propeller diameter. The velocity ratio has been changed as follows:

$$v_{te}/v_{pr} = .91,\ 1.0,\ 1.1,\ \text{and}\ 1.2 \tag{2-8}$$

Additionally, two turbine diameters have been considered:

$$D_t = 5\ \text{feet and}\ 6.5\ \text{feet} \tag{2-9}$$

With a constant propeller diameter of 18 feet, the analysis shows data for the following ratio of turbine diameter to propeller diameter:

$$D_t/D_{pr} = 0.277\ \text{and}\ 0.361 \tag{2-10}$$

The analysis has been made for propeller efficiencies of 65% and 70% and for transmission efficiencies of 85% and 90%. The data are presented in Figures 12 to 15. It is noted that for the larger values of T_{te}/T_{tot} and for the high transmission efficiencies, the efficiency of the new propulsion system can easily exceed the efficiency of the propeller alone. The study clearly indicates that larger values of T_{te}/T_{tot} and v_t/v_{pr} are beneficial

to get maximum efficiency. On the other hand, values of v_t/v_{pr} lower than one result in smaller quantities of flow being handled by the transmission, and this can be beneficial in cases where large amounts of power are being transmitted in a small transmission or where cavitation may be a problem. The data also indicate that there is a need for optimizing both maximum efficiency and low cavitation, and careful attention must be given in selecting the individual components and their performance.

H. The Low Cavitation Pump

In order to meet optimum efficiency and have a short flow path through both the pump and turbine, a pump with 50% reaction has been studied. Thus, the pump will have both inlet and exit guide vanes so that the blading will be designed for a symmetric vector flow diagram, and it has very nearly equal velocities in the impeller and the guide vanes. It is believed that such a pump will have slightly better efficiency, and there is extensive experience available regarding such designs from axial flow compressors. In addition, the 50% reaction design reduces the forces on the impeller blades and results in lower impeller thrust forces. The corresponding larger guide vanes forces will transmit thrust directly into the structure of the ship. These are all desirable features for the transmission system.

Figure 16 shows a flow vector diagram for the tip of a pump impeller for the 50% reaction pump and for a standard pump in order to indicate the reduction in flow velocity over the blades. Both pumps have a tip speed of 409 ft/sec. The relative velocity over the blades is 417 ft/sec for the standard pump whereas the relative flow velocity over the blades for the 50% reaction pump is only 236 ft/sec, or 56.6% of the velocity of the standard pump. Thus, the reduction in forces on the blades as well as the improvement in cavitation performance are substantial.

The pump with blading according to a symmetric vector diagram has distinct advantages in cavitation and reduced forces on the pump impeller blades. In addition, such a pump will result in greatly improved thrust distribution with a contra-rotating propeller. Such a pump will also permit better thrust distribution with a single rotation propeller, but the advantages are not as large. While such pumps are not in general use today, it is noted that this type of turbomachinery is widely used in axial flow compressors. The smaller blade forces will permit the use of bronze materials for the pump impeller, thus avoiding fouling of the impeller blades when the ship is in a harbor and the velocities across the blades are zero. Thus, the 50% reaction pump is of distinct advantage for use in the new hydraulic transmission.

I. Boundary Layer Control on Propeller and Ship

The hydraulic transmission will result in a turbine discharge velocity contributing to the overall thrust of the propulsive system. This discharge velocity occurring near the root of propeller blades will result in greatly improved velocity distribution in the wake of the entire propeller. It should, therefore, result in a smaller propeller with higher efficiency than is presently in use. At this time, no detailed analysis has been made regarding the quantitative benefits of this system. In addition, the suction of the pump may also be arranged to act as a boundary layer removal system at the aft end of the ship where such boundary layer removal is of greatest benefit. Consequently, the drag of the entire ship may be reduced when compared to conventional propeller installation.

The flow conditions entering the hydraulic transmission have to be carefully analyzed. However, substantial data exist for such analyses as a result of recent high speed ship propulsion systems as indicated in Kruppa (1974).

The pump of the hydraulic transmission is operating at the stern of the ship and, consequently, it will accelerate a substantial part of the ship's boundary layer. In principle, the effects on boundary layer flow are not different from existing propeller arrangements. However, the propeller does not act on the flow within the inner 20% of its radius, as mentioned earlier.

The pump of the hydraulic transmission, on the other hand, exerts a very substantial suction on the entire ship's boundary layer. As a result, with the hydraulic transmission, substantial boundary layer acceleration forces are exerted on the ship's boundary layer. This problem has not been studied in detail at this time. However, it must be considered in both the pump design and in the layout of the ship's hull. It is most likely that, from an overall systems point of view, the ship's hull should have different lines, particularly near the stern where the new propulsion system is located. Qualitatively, it can be estimated that the optimum ship's form may be a ship of shorter length and greater width near the stern.

The discharge from the transmission turbine will not only generate thrust for the propulsion system but, if properly arranged, will also provide boundary layer control near the hub of the propeller. This hub is located at a relatively larger diameter than with a conventional propeller. As a result, one can expect an increase in propeller efficiency beyond the efficiencies discussed earlier. The above effects will need experimental data to be quantified. However, the pump of the hydraulic transmission will have beneficial effects on the ship's hull and the propeller. No detailed information is available at this time regarding these benefits of the new propulsion system.

J. The Total System and Contra-Rotation

The efficiency analysis of the new propulsion system has been extended to contra-rotating propellers based on the performance data as published in Comstock (1970). In order to use Comstock's data, the performance of the propeller includes a smaller diameter ratio of D/D_{70} 0.8 and corresponding higher shaft speed ratio. At this diameter ratio, the performance of this propeller is identical to the data published by Comstock. Table 2 summarizes the data published in Comstock. The efficiency increase by going from single rotation to a contra-rotating propeller is approximately 9%, and the data indicate that for large contra-rotating propellers, efficiencies up to 85% are possible. It should be noted that with the new hydraulic transmission, the slow speed shafts and the high values of a step down ratio are easier to accomplish with a contra-rotating turbine as compared to a single rotation turbine. This fact is well known to the designers of turbines. As a result, a new transmission system offers a real opportunity for very high propulsive efficiencies with contra-rotating propellers at slow shaft speed. Figure 17 shows the efficiency of contra-rotating propellers. Additional studies of such systems are recommended in the light of the gains in efficiency which are possible with such a system. The new propulsion system eliminates the reduction gear and, as a result, there is a considerable savings in weight. For a conven-

tional gear driven propeller, there is a requirement for a slow speed shaft from the reduction gear to the propeller. This shaft not only has a large diameter, but it is frequently of considerable length, and thus of very substantial weight. These large diameter long shafts also require shaft couplings and shaft bearings of considerable size and weight. In addition, there is a shaft seal of considerable size where the propeller shaft penetrates the ship's hull. Thus, a slow speed propeller requires a substantial weight associated with the propeller shaft, bearings, couplings, and shaft seal. The new propulsion system has a high speed shaft of small diameter and associated small diameter bearings, shaft couplings, and shaft seals. It is also possible to reduce the length of the shaft because the power turbine is frequently of smaller height than the reduction gear and, therefore, can be moved farther aft.

Thus, the new propulsion system will result in considerable savings in weight and space. The entire ship can be reduced in length, can carry a higher payload, or a combination of both. The higher propulsive efficiency with contra-rotating propellers could lead to a smaller power plant, less fuel consumption, smaller fuel tanks, or a combination thereof. Thus, the new propulsion system offers a series of benefits not available with existing gear driven propellers.

IV. THE AIR MODELS

A. The Single Rotation Model

The first air model was designed and constructed to demonstrate the operating principle and the feasibility of the transmission system. In a marine application, the entire propulsion unit of the new system is located outside the ship's hull. Weight and space savings require that the turbine assembly of the transmission be incorporated within the hub of the propeller. Thus, the outer turbine casing and propeller hub rotate around the stationary, axial center shaft. The first air model was constructed so that this principle could be demonstrated. The model is easily transported and it was designed for a high speed reduction ratio, on the order of 40:1.

The power source for the air model is a variable speed, 0 to 14,000 RPM, axial flow fan which forms an integral part of the overall transmission model. The fan is die cast aluminum having four twisted blades with a chord of approximately 3.0 inches, blade height of 0.375 inches, and a blade thickness of 0.250 inches. The fan tip diameter is 3.5 inches, and its overall axial length is 1.0 inch. Directly upstream from the fan are eight inlet guide vanes which direct incoming air to the fan and support the DC electric fan motor. This DC motor has a variable speed output of 0 to 14,000 RPM, dependent upon the electrical input to the motor. The motor will function over the input range of 0 to 28 volts DC and 0 to 4.5 amperes, thus generating up to

126 watts of power or 1/6 horsepower. The fan, guide vane, and motor unit are an integral part of the model comprising 4.9 inches of its axial length.

A controllable power supply converts 110 volts, AC 60 Hz, single phase current into adjustable DC current up to 30 volts. Direct readout of current and voltage levels is obtainable from face-mounted dials as the rheostat is varied. The power supply permits continuous regulation of voltage for variable speed control of the fan from 0 to 14,000 RPM.

The turbine casing was made with an OD of 4.300 inch in order to allow a standard three-bladed Mercury "outdrive" propeller to be fitted around it. The propeller has a 16-inch tip diameter and 4.625 inch hub diameter. Within the turbine casing, there are two stages of rotor and stator blades. These are supported concentrically on a central, axial stator shaft and form the air turbine mechanism. The two rotors and two stators are each comprised of six steel blades, welded tip and hub, to steel rings. The blade profile is similar to the profile to be employed in water turbines and is untwisted with a 1.375 inch chord. These blades were received as samples from a New England manufacturing firm. Two rotor rings form the structural support for the rotating turbine casing and are mounted on ball bearings that rotate around the stator shaft.

In order to facilitate visual observance of the internal mechanisms, the housing of the entire model was constructed from clear plexiglass. Figure 18 shows a sectional view of the assembled model. The first stage stator ring is the keystone of the

model. It supports and aligns both the fan unit and turbine unit. It also houses the base support post and holds the model in position. The turbine unit is supported solely by the axial stator shaft and held in position by one bolt. The fan unit is supported by three long bolts attached to the first stage stator ring. The stator shaft seats two ball bearings upon which are located the rotor rings of the first and second turbine stages. The outer periphery of the rotor rings support the turbine's plexiglass casing over which the propeller is fitted. The two ball bearings absorb both the radial load of the rotor rings, casings, and propeller and the axial thrust forces developed by the propeller and turbine. In this model, it was not necessary to install an additional thrust bearing for the turbine casing due to the fractional horsepower involved.

Figure 19 shows a photograph of the assembled model ready for operation. Figure 20 shows a photograph of the model together with the electronic power supply. Figure 21 shows the component parts of the air transmission model.

The test of the air transmission model was entirely successful. It proved the basic principle of the hydraulic transmission and the proposed structure. It demonstrated the capability of driving a propeller with fluid energy and having the turbine completely incorporated into the propeller hub. Figure 22 shows the tested performance of the unit presenting fan impeller speed as the ordinate and turbine-propeller speed as the abscissa. It is

indicated that at the maximum speed of 14,000 RPM, there is a propeller speed of 360 RPM resulting in a speed reduction ratio of 39:1. The speed reduction ratio is not linear as indicated by the test data but increases slightly with decrease in fan impeller speed resulting in a speed reduction ratio of about 52 at 9000 RPM. This change in speed reduction ratio is caused due to the higher losses at the lower speeds. Figure 23 shows volumetric flow through the transmission as a function of impeller speed.

The test demonstrated the expected speed reduction ratio together with the integral design of turbine and propeller. It also clearly showed the advantages of having the propeller decoupled and not rigidly connected to the driving unit. The test of the air model transmission and propeller unit met all requirements.

B. The Contra-Rotating Model and Propulsion System

In order to demonstrate the simplicity of contra rotation in connection with the new propulsion system for ships, an air model has been designed and constructed. The basic design of this model is shown in Figure 24.

An integral DC electric motor and single rotation axial flow impeller form the power source of the model. The DC motor has only low fractional horsepower, a maximum of 1/10 HP to the impeller. The precision cast, four-bladed, aluminum impeller has a speed range of 0 to 16,000 RPM, and impeller speed is a fairly linear function of the input voltage (0 to 37 volts). The four twisted

impeller blades have a cord of 1.125 inches, height of 0.5 inches, thickness of 0.150 inches, tip diameter of 2.700 inches and axial length of 0.700 inches. Directly downstream from the impeller are three exit guide vanes which direct the air, via the transition section of the transmission, into the first stage stator ring of the right hand propeller.

A controllable power supply converts 110 volt, 60 Hz, single phase current into variable DC current up to 40 volts. Direct current and voltage readings are obtainable from face-mounted dials as the rheostat is varied. The power supply permits continuous regulation of voltage for variable speed control of the impeller from 0 to 16,000 RPM.

The turbine is of the contra-rotating type and is of the inverted turbine construction. Two propellers operate in conjunction and contra rotation. The second turbine discharges to atmosphere, and this discharge flow makes an appropriate contribution to the overall thrust.

The turbine casings have an OD of 4.250 inches in order to allow standard three-bladed, aluminum, Mercury Company "outdrive" propellers to be fitted over them. Two standard Mercury Company propellers, identical except for opposite rotation, having a tip diameter of 15.0 inches and a hub diameter of 4.625 inches were employed as the contra-rotating propellers. Within the hub of each propeller is contained two stages of rotor and stator blades. The unit is designed so that the forward propeller rotates clockwise (right hand) and the aft propeller rotates counterclockwise

(left hand). The rotor and stator assemblies each contain ten non-twisted steel turbine blades with a 0.625 inch cord. The two rotor rings contained in each hub form the structural support for the hub and transmit rotational energy to the propellers. The rotor rings maintain radial and axial stability from ball bearings that are mounted on the hub of the second stage stator assembly in each right hand and left hand unit.

Figure 24 shows a sectional view of the contra-rotating model. The first stage stator ring is the keystone of the model. It supports and aligns the drive motor, impeller, forward housing and both right hand and left hand turbine units. It also houses the base support post and holds the model in position. Both right hand and left hand turbine units are supported solely by the axial stator shaft and held in position by one bolt. A clear acrylic annulus surrounds the impeller to allow for visual observation. Electrical lines are concealed within the support post so no wires are visible external to the model. Bayonet connections allow the support post to be separated from the base plate without having to disconnect any wires for model disassembly for transport.

Figure 25 shows a photograph of the assembled model ready for operation. All of the photographs were taken prior to black oxide treatment of steel components and anodizing of aluminum components, which imparts a uniform black finish on the metal surfaces. This treatment improves the visual appearance of the model and prevents deterioration of the components due to corrosion. Figure 26 shows the component parts of the contra-rotating air transmission model.

Figure 27a shows the performance of the contra-rotating model presenting impeller speed versus right hand and left hand propeller speed. It is indicated that at the maximum impeller speed of 16,000 RPM the right hand propeller speed was 280 RPM and the left hand propeller speed was 230 RPM.

This results in a speed reduction ratio for the forward, right hand propeller of 57:1 while the aft, left hand propeller has a ratio of 69:1. At this impeller speed, 16,000 RPM, the aft, left hand propeller runs approximately 18% slower than the forward, right hand propeller. This percentage decreases as impeller speed decreases and the aft propeller demonstrates an average of 13% slower speed than the forward propeller. The speed reduction ratio of both propellers increase as impeller speed is reduced. This is primarily due to system losses being highly reflected as a result of the low fractional horsepower drive unit of the model. If both contra-rotating propellers have identical blade configurations, except for right hand and left hand arrangement, then the aft propeller must operate at slightly lower shaft speed to transmit the same power as the forward propeller. This effect is caused because the fluid enters the aft propeller with a predetermined rotational flow component. The amount of rotational speed difference can be selected by appropriate blade angles in the aft turbine. By using adjustable turbine nozzles, the propeller speed and the speed difference between forward and aft propeller can be changed over a wide range. Figure 27b shows horsepower and input voltage as a function of impeller speed. With contra

rotation, high transmission ratios can be achieved much more efficiently. With the increased importance on overall efficiency and low fuel consumption, contra rotations will find many more applications in the future. By using a single rotation high speed impeller driven directly from the prime mover, only a single seal is required. Since the impeller operates at relatively high speed, the shaft, bearings, and seal have relatively small dimensions. The model has been built to demonstrate the simplicity of contra rotation in conjunction with the new propulsion system. In the model, each turbine has two stages resulting in four turbine stages for the entire model. The mechanical design and the number of bearings can remain unchanged even for larger and highly efficient water turbine units, with each turbine having many stages. The model may serve as a prototype for the mechanical design of a much larger water propulsion unit.

V. THE WATER MODEL

A. Pump Design

In order to prove the concept of the hydraulic transmission, a water model was designed and built. In analyzing the hydraulic transmission model, a number of pump and turbine combinations have been studied, resulting in pump performance covering a range of pressures, flows, and specific speeds. In these studies, both conventional pumps with an unsymmetric vector diagram as well as low cavitation pumps with a symmetric vector diagram have been considered. For the hydraulic transmission, the efficiency is most important so that the new propulsion system will exceed the efficiency of existing systems. The efficiency of a hydraulic transmission depends on the individual efficiency of pump and turbine. However, the efficiency of pumps and turbines decreases with an increase in specific speed. Thus, the pump efficiency is the critical element for the total transmission efficiency because the pump operates at very high specific speed.

The objective of the water model was to demonstrate the basic principle of the hydraulic transmission and to get experimental data of potential efficiencies. Another objective is the analysis and verification of cavitation characteristics of

such a transmission. Published data indicate higher efficiencies for large quantities of flows, but they show a significant reduction in efficiency with increasing specific speed, as shown in Figure 28. The use of laminar flow airfoil blades and improvements in technology results in increases in efficiencies which has been estimated and is shown as a dotted line in Figure 28 for pumps of large capacity and high Reynolds numbers. The pump for the experimental transmission will operate at considerably smaller power input, capacity, and Reynolds number. Thus, careful consideration must be given to specific speed, flow quantity, Reynolds number, and other factors in the selection of the pump specification in order to achieve the desired high efficiency.

Another consideration in the selection of the pump is the effects of cavitation. Cavitation performance is generally defined by the suction specific speed, and the values for suction specific speed must be below a value of n_{ss} = 10,000 for small pumps. With the above consideration in mind, the detailed specifications for the pump design were developed and are shown in Table 3. The pump of the experimental hydraulic transmission has a relatively small capacity with a range of Q = 1032 gal/min at 7500 RPM and Q_1 = 619 gal/min at 4500 RPM. The design specific speed of n_s = 0.541 of N_s = 9302 is very high for a small

pump. The pump has been designed with conventional blading, having an unsymmetric vector diagram, since cavitation and blade stresses are not expected to be a problem in the small size, and it was desirable to prove a good efficiency before introducing the additional innovation of a 50% reaction pump.

The pump has been designed with fixed blades. In studying the hydraulic transmission, it became evident that a pump with adjustable blades could result in a hydraulic transmission with a variable transmission ratio. In order to simulate in the model the performance of adjustable pump impeller blades, it was decided to design and build a second pump. In this pump, only the impeller is changed, and within the impeller, all dimensions are identical to the first pump except the stagger angle is 5° less than the first pump. Thus, it will be possible to test the transmission under conditions of about 20% less flow and an appropriate reduction in pressure.

B. Pump Manufacturing

The pump blades are NACA 65 family laminar flow blades for which mathematical equations defining the detailed blade contours are available. The entire pump impeller and the guide vanes were manufactured on a three axis tape controlled milling machine. All airfoil sections of both the guide vanes and the impeller vanes were machined by this method. The NACA airfoil data were

programmed on the URI IBM computer, and the data were transferred to the milling machine tape. The entire impeller was constructed according to this method with some rough machining before the milling operation. After completion of the milling operation, the blades were hand finished. The guide vanes were manufactured by the same method and subsequently silver soldered into concentric hub and tip rings. The hub ring positions and holds the discharge diffuser rigidly in place. The details of the pump design, manufacturing, and testing is described in Sheets (1979-1). The complete pump design and drive system is shown in Figure 29, and the pump components are shown in Figure 30. The complete pump assembly and transfer case is shown in Figure 31.

C. Turbine Design

Since the turbine operates in a flow field of accelerating flow, many design decisions are less critical as compared to the pump. Among the large number of variables are hub to tip ratio, total pressure, which determines the number of stages, flow and flow coefficients, head per stage, and the ratio of circumferential blade velocity, u, to absolute flow velocity, c. The design analysis was started by selecting 50% reaction for both stationary and rotating stages. Within this requirement, the available head resulted in approximately 40 to 45 degrees

deflection in both stationary and rotating stages for a certain range of turbine diameters. A number of configurations were analyzed which met the basic turbine requirements. The initial studies indicated that a turbine with 6.0 inches tip diameter and 4.0 inches hub diameter would meet the flow and pressure developed by the pump. The number of stages, while a variable, was on the order of seven or eight stages. In these studies, the design objective was the lowest possible meridian velocity and associated low water velocity across the blades, as this would result in optimum turbine efficiency. A number of turbines were analyzed on this basis with the objective of optimizing for low flow velocity and minimum number of stages within the range of required turbine diameters. Basically, these turbine designs had a meridian velocity of about 0.6 of the equivalent pump velocity. It was expected that the flow transition between pump and turbine would accommodate this flow deceleration. During the design studies, it developed that the installation of the flow reversing mechanism in the transition area between pump and turbine did not permit the above mentioned amount of flow deceleration and simultaneously have pump and turbine located closely to each other. It was, therefore, decided to have a smaller flow deceleration between pump and turbine resulting in a value of meridian velocity for the turbine equalling about 0.9 of the equivalent pump value. This, in turn, resulted in a

small reduction of the turbine tip diameter and a small increase of the turbine hub diameter in order to reduce the turbine annulus area for the desired flow values. This modification resulted in higher flow velocities, a higher flow coefficient, and also a higher pressure coefficient. Due to the very low turbine speed of 750 RPM, all flow velocities are low, and a slight increase in water velocity over the blades should not affect the total efficiency.

A number of studies were made to determine the optimum turbine configurations, and for the turbines, straight blades from hub to tip were selected since these were the only blades available. A manufacturer of turbine blades made available several sections of extruded turbine blades. These turbine blades were modified from the original extruded blade contours to get a turbine blade profile of lower drag and modified camber line in order to get high turbine efficiency. As a result of these studies, turbine number 23 and 24 of Table 4 have been selected for design and construction. Turbine 23 and 24 are identical except that turbine 23 has five stages and turbine 24 has four stages. The turbines have a tip diameter of D_{Tt} = 5.7 inches and a hub diameter of D_{Th} = 4.4 inches. The basic design data for turbine 23 and 24 are shown in Table 4, and the blades for nozzles and rotor are identical. Table 5 shows the summary of the turbine calculation data which are the basis for the turbine

blade design shown in Figure 32. The calculated data of Table 5 and Figure 32 show values according to the free vortex theory. Since the turbine blades are untwisted, the values at the mean diameter are controlling and the values at tip and hub will be determined from the single stage air turbine test. The calculated flow vector diagrams are shown in Figure 33 for turbine 23-T-7-3 and in Figure 34 for turbine 23-T-7-4.

D. Blade Cascade Tests

Since a multistage axial flow water turbine had never been built, it was decided to make blade cascade tests in air and build a single stage air turbine before building the multistage axial flow water turbine. The cascade tests have been designed to establish flow turning data and information concerning the total pressure loss across the turbine blade rows at various Reynolds numbers. This information will serve as a basis for the design of the air turbine and the water turbine. Valuable insight could thus be gained as to the correlation between air turbine test results and predicted water turbine performance. The cascade tests were made using air, and they will give test data with untwisted blades. Three groups of cascades were tested simulating different stators. Each cascade was tested at three different air velocities, thus testing at different Reynolds numbers. The stagger angles incorporated into the

three cascades were: 26.9 degrees, 20.8 degrees, and 16.5 degrees. These angles also become the angles of attack (α) if the inlet flow is axial ($\beta_1 = 0$). This inlet condition was selected throughout the cascade tests. The blade configuration for the three groups of blades is summarized in Table 6 with values of blade angles at hub, mean, and tip of the straight blades. The tip diameter is 6.25 inches, and the hub diameter is 4.00 inches.

Figure 35 shows an arrangement of the cascade tests. The data from the individual cascade stator tests is summarized in Table 7. The table shows the cascade nozzle efficiency, which is a measure of exit total pressure to the inlet total pressure. The measured and calculated exit velocities are also presented. These values compare within 5% and, therefore, indicate the reliability of the test data. As the table indicated, three stators were tested. The test data show good nozzle efficiencies with a maximum efficiency of over 94%, and the deflection angle is quite satisfactory, particularly for the stator number 5. The fluid outlet angle is shown in Figure 36, and it indicates a straight line performance. The results of these tests were used in designing the air turbine and, subsequently, the water turbine.

E. Air Turbine Tests

Since the water turbine is of unconventional design, being a multistage axial flow turbine of the inverted type with the rotating blades connected to an outer rotating arm, it was decided to design and build a single stage air turbine using the same blading which is used in the water turbine. This air turbine is of conventional design having a shaft which supports a disk which in turn carries the blades. This turbine had a tip diameter of 6.25 inches and a hub diameter of 4.00 inches. The turbine shaft drives a dynamometer to measure the power output of the turbine. The objective of the testing is the determination of flow deflection created by both the rotating and stationary blades. In addition, power output and efficiency will be determined. Since the single stage test uses air as the fluid, the turbine must operate at substantially higher speed than the planned 750 RPM of the water turbine in order to operate at a similar Reynolds number. The fluid energy for this turbine is provided by an electrically driven axial flow fan which is available in the Mechanical Engineering laboratory. This test required measurement of air pressure, flow, shaft speed, and power output. In addition, flow deflection will be measured. The air turbine also permits variation in the axial distance between the stationary inlet vanes and the rotating turbine in an effort to optimize efficiency as a function of axial spacing.

In the design of the turbine, efficiency has been compromised by using the available blades and having a straight blade without twist and change in chord length. With 265 blades in the five stage water turbine, considerable time and cost has been saved by using the available blades. It is recognized that for optimum efficiency, the turbine blades should have a certain amount of twist from inner to outer diameter and also a change in chord length may be desirable. For maximum efficiency, a lower water velocity should have been selected through the turbine and over the turbine blades. The local flow velocities over the turbine blades are about one half of the equivalent water velocity over the pump blades. The single stage turbine was tested in the laboratory to determine total efficiency and overall performance. For this purpose, three blade settings in both stator and rotor were tested, covering a range of flows and Reynolds number. The change in Reynolds number was established by operating the turbine over a range of shaft speeds. Figure 37 shows an arrangement of the air turbine test. The design point efficiency for the three turbine configurations is tabulated in Table 8 for design point flow. In the table, the turbine designation has as the first number the turbine blading, and as the second number the nozzle or stator blading. The last two numbers multiplied by 100 are the turbine speed. The table demonstrates that efficiency is dependent upon Reynolds

number. For a given stator-rotor combination, the efficiency increases as Reynolds number is increased, which in turn, is accomplished by an increase in rotor speed. Efficiency for a turbine is the ratio of power output over energy input. In the air turbine analysis, power input is determined from the air flow rate and pressure and power output is measured by the dynamometer. For a constant rate of flow, the efficiency is equal to the pressure ratio.

$$\eta_T = \frac{\Delta P_{expt}}{\Delta P_{th}}$$

The maximum turbine efficiency is well over 90% and is remarkably high for a unit of this small size and operating at low Reynolds number in addition to the various compromises in design which had to be made. For the turbine combination 75, the tested performance of the air turbine over a range of pressures and flow is shown in Figure 38 in terms of nondimensional flow coefficient, ϕ, and pressure coefficient, Ψ, together with the theoretical performance which is a straight line. The point of maximum efficiency is presented on the theoretical line by x and the actual tested maximum efficiency is the nearest test point at 3800 RPM. The efficiency of the water turbine may be slightly lower because the air turbine test is essentially a stage efficiency and does not include inlet and exit losses.

In addition, the water turbine has a higher hub to tip ratio due to the requirements of the reversing mechanism in the water model. On the other hand, the Reynolds number in the water turbine will be above the Reynolds number for the air turbine. With the above test data, there is enough technical information to design a five stage water turbine of high efficiency.

F. Water Facility and Testing

The pump test system is of closed loop design, using six inch schedule 40 PVC pipe for the flow loop and a modified decompression chamber as the fluid reservoir chamber. With a capacity of approximately 1300 gallons, the fluid chamber serves as both a limited settling chamber and reservoir for the pump. Restricted space considerations and the lack of availability of a larger reservoir capable of pressurization precluded the use of a larger chamber, which would have been desirable in order to avoid the possibility that flow turbulence in the test chamber might adversely affect the pump inlet conditions. It should be noted that the flow loop is arranged so that turbulence caused by the flow entering the fluid chamber will have a maximal chance to dissipate before passing into the pump inlet, see Figure 39. Facilities and equipment for a quantitative analysis of the effects of any possible turbulence in the pump inlet were not available. However, any turbulent flow effects would be detri-

mental to the overall pump performance, and as such the results obtained under these conditions are conservative relative to ideal conditions.

Pump cavitation testing led to the incorporation of a pressurization system for the test chamber. Modification to the test chamber, originally a decompression chamber with a rated working pressure of 100 psig, indicated that a maximum working pressure of approximately 25 psi can be maintained. The main air supply has a pressure of 125 psi, and the pressurization system is controlled by a three-way valve and a Schrader five to 125 psi regulator. The gross regulated pressure was monitored with a Bourdon tube dial gauge. Since the local pressure at the pump inlet is the gross regulated pressure plus the local hydrostatic head, an independent system, to be described later, is used to more accurately monitor the inlet pressure.

In order to avoid the possibility of corrosion, it is important that the pump and associated system elements be removed from the water when testing is not taking place. The water is treated with a softening agent to ensure that its Ph is maintained slightly basic. Refilling the test chamber with fresh water and treating that water before each test session is unacceptable in terms of both cost and time. Therefore, a holding tank and associated equipment is included in the test chamber system. The holding tank has a capacity of approximately 2200

gallons. The water is transferred between the holding tank and the test chamber with a TEEL model 1P851 centrifugal pump rated at 130 gallons per minute maximum flow. A six inch BIF model 6B3.610 universal venturi tube is mounted in the test system flow loop in order to monitor the flow generated by the pump. Two methods of monitoring the pressure difference across the venturi tube are used. A mechanical differential pressure gauge and a 100 inch manometer are connected in a parallel arrangement. The Barton model 227-109608 bellows type differential pressure gauge with a 0 to 100 inches of water range was installed. Because the maximum pressure difference, experienced at the maximum flow rate, exceeds the 100 inch range of the Barton instrument, a Merriam model M-100 100 inch manometer was used for high flow monitoring. With manometer fluid of specific gravity 2.94, this manometer allows readings up to 194 inches of water, which covers the range of interest, and is accurate to approximately 0.2 inches of water.

Multiple pressure taps are used to measure pressure at various stations of interest in the pump and in the turbine. The complexity of the pump housing forced the use of three pressure taps per station instead of the ASME recommended four. To help ensure uniformity of results, a single Bourns 0 to 150 psi pressure transducer measures the pressure at each tap. A valve manifold system is used to allow the pressures to be applied

individually and in an organized manner to the transducer. The output of the transducer is read on a Hewlett-Packard 3467 B digital multimeter. A General Radio Company type 1531-B Strobotac with a range of 110 to 25,000 RPM is one method used to measure the pump's rotational speed. The Strobotac triggered a Hewlett-Packard model 523C electronic counter which provided an accurate digital measure of the pump's RPM. A Servotek type SB740AZ generator tachometer is also used to measure the pump's RPM. It is connected to the hydraulic motor in the transfer case and its output is connected to a Newport model 204-3 D1 digital voltmeter for readout.

The input power to the hydraulic motor is monitored by measuring both the hydraulic fluid flow rate and its pressure. The flow rate is measured with a Fischer and Porter Flowrator-type rotameter. This two tube rotameter has a small tube with a range of 0.92 to 5.1 gallons per minute and a large tube with a range of 4.7 to 25.5 gallons per minute. The hydraulic pressure is measured in two ways -- with a pressure transducer whose output is read with the Hewlett-Packard digital multimeter, and with an Ashcroft 0 to 6000 psi Bourdon tube dial gauge. Electronic temperature probes are used to monitor the hydraulic fluid temperature and a water-cooled heat exchanger was installed in the hydraulic test stand to ensure that the fluid temperature remains within acceptable bounds, i.e., less than 50^{o}C.

The arrangement of the entire test loop together with the test chamber, holding tank, and the hydraulic power generation unit is shown in Figure 40. Figure 41 shows the pump installed in the test loop with the instrumentation and pressure taps previously described. In this photograph, one can see the pump blade through the plexiglass part of the pump housing.

The turbine and transmission test is done in the same facility with the same instrumentation. There is a slight modification in the test loop, and Figure 39 indicates the configuration of the test loop for transmission testing.

G. Pump Tests

The pump is the most important component of the transmission and propulsion system, and it is a critical element in the transmission efficiency. Two pumps were tested and separate reports have been issued for each pump test, Sheets (1979-1) and Appendix A. The pump testing arrangement is shown in Figures 39 and 41. The pump is driven by a shaft which is powered by a hydraulic gear motor through a 1:2 setup ratio within the transfer case. The hydraulic motor can deliver various horsepower, up to 22.0 horsepower depending on line pressure and flow, at speeds of 0 to 4000 RPM. Thus, when the motor is the driver in a 1:2 ratio, pump output speeds range from 0 to 8000 RPM. A clear acrylic annulus was incorporated in the pump casing

surrounding the pump impeller. This feature allows visual observation of the impeller during all pump testing by synchronizing impeller speed with an external strobe light tachometer. The impeller is "stopped" at all test speeds with the strobe light and any visual signs of bubble development as a result of dissolved gas or cavitation are recorded.

Pump testing for both pumps is carried out at 1500, 2000, 2500, 3000, 4500, 5000, and 6000 RPM. At each speed, tests begin at free flow and subsequently, the flow rate is throttled to obtain sufficient data to generate performance curves from free flow to shutoff. Input horsepower to the transfer case, pump flow rate, pump speed, and various pressures from pressure taps are monitored and recorded. The performance curve of the pump 1 is shown in Figure 42, and the summary of the efficiency data is presented in Table 9. It is noted that the total efficiency reaches 90.9% with the maximum stage efficiency reaching 94.4% and the maximum system efficiency being 82.2%. At 4500 RPM, all maximum efficiencies are somewhat lower, namely the stage efficiency equals 87.2%, the total efficiency being 86.0%, and the system efficiency being 74.2%. At 5000 RPM, maximum efficiency is again lower with the stage efficiency being 85.7%, total efficiency being 84.0%, and system efficiency being 77.0%. The data at 6000 RPM are somewhat inconclusive, but efficiencies well above 80% have been recorded for some

conditions.

The total efficiency for pump 2 is shown in Figure 43 with a maximum value of 84.1%, and the stage efficiency has a maximum value of 84.8%, both at 4500 RPM as shown in Figure 44. The efficiencies for pump 2 are higher at 3000 RPM, but they have not been quoted since the accuracy at the low power inputs was not within the required tolerances. The characteristic of both pump 1 and pump 2 in terms of nondimensional flow coefficient, ϕ, and pressure coefficient, Ψ, is shown in Figure 45, and the maximum efficiency of both pumps is presented in Table 10. For use in the transmission, the stage efficiency should be used in calculating the total transmission efficiency. In considering pump 2, it is recognized that due to an error in balancing and associated repair, lower efficiencies have resulted.

The most noticeable result is the remarkably high efficiency and the fact that efficiency decreases slightly with increased speed. A careful study of the data has indicated that the small reduction in efficiency with increased shaft speed is most likely caused by the test arrangement and measurement. The settling chamber has a capacity of 1300 gallons. At 6000 RPM, the pump will empty the entire settling chamber in 1.57 minutes. However, since there are no baffles inside the settling chamber and the pump inlet and discharge pipe are located on the center of the settling chamber and on the same side, it can be assumed

that the water movements inside the settling chamber will relate primarily to that quarter of the settling chamber which is located near the inlet and exit ports. This part of the settling chamber will be emptied in 23.5 seconds at 6000 RPM. Thus, it can be expected that there exists considerable turbulence inside the tank, and this turbulence will increase with an increase in flow or speed and reduce with a reduction in flow or speed. Under these circumstances, the static pressure measurement inside the tank should no longer be used to indicate total pressure, and the indicated efficiency may be lower due to this pressure measurement.

The cavitation characteristics were in line with the predicted performance. Pump 1 operated cavitation free to a speed of 4500 RPM over a substantial part of the pump characteristic. At a speed of 5000 RPM, cavitation free operation was observed in a narrow range near the design point. The test data indicate that the efficiency of both small four inch diameter pumps using laminar flow blades is very high and values up to 94.8% have been measured for the stage efficiency. Thus, the entire water model transmission can be expected to have good efficiency.

H. Transmission Tests

The detailed report on the "Hydraulic Transmission Test Program" is presented in Appendix B covering transmission de-

sign and specification, transmission fabrication, transmission test system, and the transmission tests. In this section, a summary of the test data is presented, and the total turbine and transmission performance is discussed. The test setup, including the flow loop and test chamber, is shown in Figure 46. Some of the details of the turbine installation and instrumentation are shown in Figure 47. This test setup proved quite satisfactory to perform the turbine tests in accordance with ASME turbine test specifications. The same test setup was used for the complete transmission test, and it gave results with sufficient accuracy. The test arrangement had some limitation in the length of the piping of the flow loop and the size of the test chamber. These limitations had been previously discussed in the pump test program.

The testing of the unit consisted of a turbine test, in accordance with ASME test procedures, which was followed by a complete transmission test. The turbine tests were made with a pump speed range from 3000 to 5000 RPM covering turbine speeds on the order of 450 to 1000 RPM. The turbine was tested both in the four and five stage configuration. The transmission test covered both the four and five stage turbine together with two turbine diffuser configurations using both the short and long pump guide vanes.

Transmission tests were conducted over a period of five months in which testing was evaluated, and in some cases, repeated to ensure the reliability of the data. Results from testing under the following conditions are presented:

DATE	PUMP	NO. OF TURBINE STAGES	TYPE OF DIFFUSER	TYPE OF PUMP EXIT GUIDE VANES	TEST DATA (FIGURE NO.)
4/26/79	1	5	Bullet	Long	48
5/31/79	1	5	Improved	Long	49
6/1/79	1	5	Bullet	Long	Appendix B
6/12/79	1	5	Bullet	Short	Appendix B
8/21/79	2	5	Bullet	Long	50
10/5/79	1	5	Bullet	Long	Appendix B

There are two main goals of the transmission tests. The first is to evaluate the performance of the new "inside-out" multistage axial flow turbine. Second is the evaluation of the transmission system's performance in order to prove or disprove the feasibility of the concept. Additionally, comparison of tests with pump 1 and pump 2 will indicate the transmission performance changes which can be achieved with controllable pitch pump impeller blading.

Figure 48 shows the performance of the five stage turbine with the long exit guide vanes, the bullet diffuser, and pump.

1. The figure shows turbine flow coefficient as abscissa and turbine total efficiency as the ordinate. The data indicate an increase in turbine efficiency with larger flow rates due to higher pump speeds. The larger flow rates result in a higher turbine Reynolds number and, therefore, the performance increase to higher turbine efficiencies with higher pump speed is to be expected. It is noted that the turbine bearing losses are included in the total turbine efficiency. Figure 49 shows the turbine total efficiency as a function of turbine flow coefficient when testing the same turbine with the improved diffuser instead of the bullet diffuser. It is noted that the performance is quite similar, but the improvements in turbine performance with higher pump speeds is less pronounced. Figure 50 shows the turbine total efficiency as a function of turbine flow coefficient when testing pump 2 with the same turbine configuration. It is noted that the turbine performance with pump 2 shows a slightly lower total efficiency, and this performance is caused by the different flow and pressure generation of pump 2 as compared to pump 1. Nevertheless, the turbine performance is still excellent, reaching efficiencies up to 85%. Figure 51 shows the five stage turbine performance in terms of turbine efficiency, turbine head, and turbine output power as a function of turbine flow with turbine shaft speed as independent variable. The data show that the turbine covers a range between

one half and seven horsepower and that the turbine can cover a wide range of flow and head conditions with excellent efficiency, namely between 85% and 90%, provided the pump can deliver flow and head in these specific ranges. It is also an indication that the five stage axial flow water turbine can operate over a wide range of operating conditions at very high efficiencies.

Figure 52 shows the transmission efficiency as a function of transmission ratio with pump speed as independent variable. The same data are shown in tabular form in Table 11. A maximum transmission efficiency of 65% is shown at the lowest pump speed of 3000 RPM. Table 11 indicates that turbine efficiency significantly increases with pump speed but that the pump efficiency decreases with increasing pump speed. There is a slight mismatch between turbine and pump so that the pump does not operate at its point of maximum efficiency. In addition, the pressurization of the test chamber had been adjusted to have cavitation free operation near the design point of the pump. Under the test condition for maximum transmission efficiency, it is indicated that a higher amount of pressurization would have been needed to keep the pump efficiency at its maximum value.

Figure 53 and Table 12 show the same data for the transmission when tested with the improved turbine diffuser. It is noted that at the lowest pump speed the turbine shows a higher

efficiency and as a result, the transmission efficiency is slightly higher than the tests show with the bullet diffuser. However, this improvement in turbine efficiency is not maintained at the higher speeds. The previously made comments regarding tank pressurization are applicable to this set of tests as well. It is noted that the transmission tests with the improved diffuser show a slightly higher turbine flow coefficient and turbine stage head coefficient. Figure 54 and Table 13 show the transmission test using the bullet diffuser and pump 2. This combination of pump and turbine shows the highest transmission efficiency due to a better matching between pump 2 and the five stage turbine. While in this combination, the turbine efficiency is slightly reduced, and the power output is also slightly reduced, the total efficiency is increased primarily because the pump efficiency is increased. Pump 2 operates in a range of flow coefficients which have higher pump efficiency. Otherwise, the performance is similar to the previous test data showing a decrease of transmission efficiency with increased pump speed. It is noted that there is a significant increase in transmission ratio with pump 2.

A summary of all transmission tests is shown in Table 14 which shows peak transmission efficiency for all tests using both pump 1 and pump 2. In these data, only the pressurized performance is shown in the table for the higher pump speeds.

It is also noted that the maximum transmission efficiency of 71% occurs with pump 2 at 3000 RPM and a transmission ratio of 7:1.

Test data indicate that the model hydraulic transmission operates with 71% maximum efficiency. The efficiency exceeds the 50% to 60% value predicted in earlier studies. The 71% efficiency is an excellent value considering the many innovations which have been made and the small size of the model test unit. With enough time and funds, the small water model unit could be modified to match pump and turbine performance so that total efficiencies of 80% could be achieved. The innovations of the tested unit include the open hydraulic transmission, incorporation of a reversing system, and unusual turbine design and thrust bearings. Conclusions of these tests indicate that a multistage axial flow water turbine of the inverted type can have efficiencies of over 90% in moderate sizes and powers. The high turbine efficiency confirms the concept of an "inside-out" turbine and indicates that the rotating outer housing has little impact on turbine performance and efficiency. The tests have successfully completed the proof of concept and have shown a higher efficiency than was expected. The tests have also proven that axial flow hydraulic transmissions can be built in small sizes to fit into the hub of ship propellers.

I. Acoustic Tests

The acoustic performance of the hydraulic transmission is of considerable interest. In order to get some estimate regarding the potential acoustic noise generation of this transmission, tests were made by installing a hydrophone in the water tank at a location close to the pump inlet. The hydrophone was located inside the tank near the pump inlet pipe and laterally near the center of the tank. At this location, the hydrophone was located approximately three feet from the pump impeller and approximately the same distance from the hydraulic motor which was located inside the tank.

The spectral analysis of the experimental transmission noise was determined using the following instrumentation. Waterborne sound associated with the testing of the experimental transmission was recorded using a calibrated hydrophone, an instrumentation amplifier, and a laboratory quality tape recorder. After the test program, the recorded data were analyzed using the recorder and a sound and vibration analyzer. The analyzer was manually tuned and its output tabulated. The tabulated data are presented as power spectra.

The measurement system is shown schematically in Figure 55. Sound in the tank containing the experimental apparatus was picked up by the Clevite CH-24 hydrophone, amplified by the Ithaco 253A amplifier and recorded on the Hewlett-Packard 3960A

recorder. The frequency response of the hydrophone is shown in Figure 56. Both the amplifier and the recorder have 3dB bandwidths of at least 30 kHz, so no frequency dependent amplitude corrections are necessary. The oscilloscope was used during all test runs to verify that amplitude limiting did not occur at any time.

The frequency analysis system is shown in Figure 57. After the desired number of mechanical test runs were completed, the recorded data were replayed and applied to the analyzer. Preliminary scans were made with the analyzer to determine the bandwidth of the noise signal. Based upon the results of the preliminary scans, a measurement bandwidth was selected and sufficient noise power measurements were made to characterize the distribution of noise power over the selected frequency range.

Figures 58 through 62 show the results in terms of noise power versus frequency as measured by the wave analyzer. When operated in the "narrow" or "one third octave" modes, the analyzer bandwidth increases with the center frequency to maintain either the one third octave bandwidth to the -3dB response points or the "narrow" bandwidth of ±4% of the center frequency to the -3dB response points. To derive the true spectral density level, expressed in terms of noise energy per one Hz frequency band, a correction must be applied to the data to compensate for the increase of bandwidth with center frequency.

The tests were made with a pump impeller speed between 2000 and 6000 RPM. At these shaft speeds, the dominant pump frequencies with the four-bladed impeller are in the range of 133 Hz at 2000 RPM to 400 Hz at the maximum speed of 6000 RPM. The dominant turbine frequency is on the order of 12% lower than the pump frequencies considering that the turbine speed is on the order of 1/8 of the pump speed, and the turbine has 28 blades. The tests were made with the pump in operation without driving the turbine. The experimental data presented in Figures 58 to 62 show the acoustic measurements for 2000, 3000, 4500, 5000, and 6000 RPM. The objective of the tests was the determination of any potential large noise emission from the transmission pump. It is realized that the tank had a rather small volume of water, and it presented a steel shell with steel enclosures. Thus, the entire acoustic measurement was made in a very reverberant environment.

The test data at 2000, 2500, and 3000 RPM indicate that the measured acoustic value at the resonant frequency of 133 and 200 Hz, respectively, of the pump impeller is relatively low, and there is substantially more noise in a higher frequency range between 600 and 2000 Hz. This frequency range is in the domain of the hydraulic motor driving the pump. At 4500 RPM, as shown in Figure 60, the general noise level is on a higher plateau. However, it is noted again that the prime pump

noise emission at 300 Hz is substantially lower than the noise generated in the 800 to 2400 Hz range which covers the noise emission of the hydraulic motor. The tests were made over a wide range of flow conditions, and in the frequency range of the pump noise emission, it is noted that larger noise values exist for the lower flow quantities. At these lower flow quantities, the pump blades are in partial stall and, therefore, are likely to be the cause of higher noise levels. It is noted that at the optimum pump efficiency, the measured noise in the main frequency range of the pump impeller is considerably lower. On the other hand, the noise emission in the very high frequency range on the order of 5000 to 10,000 Hz and above, is independent of the flow and horsepower input into the pump. At 4500 RPM, as shown in Figure 60, there are much higher sound levels in the frequency range below 200 Hz. This increase in sound level may be caused by a hydraulic resonance as the sound level is greatly reduced at 5000 RPM, as shown in Figure 61. Otherwise, the tests at 5000 and 6000 RPM, Figures 61 and 62, show a similar performance to the 4500 RPM test. However, all noise levels are relatively higher.

The above tests give only very limited information regarding acoustic performance because the specific test setup did not lend itself to make a scientific acoustic analysis. The information gained from this test indicates that the hydraulic motor emits

considerably more noise than the pump of the hydraulic transmission. This fact is remarkable because the water flowing through the pump is directly connected to the tank and the hydrophone, but the fluid of the hydraulic motor is in a closed circuit and never connects directly to the water in the tank. It is noted that in an actual installation, the main pump noise frequency will be substantially higher than the propeller noise frequency. Thus, any pump noise radiation will attenuate much faster with distance than the propeller noise.

J. System Analysis

The water model has been designed to include a thrust reversing system which has been tested. The reverse thrust mechanism for the axial-flow hydraulic transmission is shown in Figure 63. In the forward thrust mode, air pressure through port B locks the reversing ring into its seat in the exit guide vane support. In this mode, the reversing ring forms the outer wall of the pump to turbine diffuser. In the reverse thrust mode, pressure to port B is vented, and air pressure is supplied to port A. The resulting force on the actuating pistons pushes the reversing ring aft onto its seat on the shoulder of the turbine stator hub. The pump flow is diverted through nozzles machined in the exit guide vane support, creating the hydro-jet which produces the reverse thrust.

Testing of this reversing system was conducted on November 12, 1979. A schematic of the reversing mechanism actuating system is shown in Figure 64. A total pressure Keil probe was installed in one of the reversing nozzles. Measurements were taken of test stand flow and pressure and pump speed. Pump flow, the settling chamber total pressure and the total pressure in the reversing nozzle were also monitored. Note was also taken of the time and pneumatic pressure required to actuate and de-actuate the reversing mechanism.

In all cases, the time required for the reversing ring to seat after it began moving was much less than one second. If insufficient pneumatic pressure was applied, however, the mechanism would not actuate. The pressure required to actuate the mechanism increased with increasing flow rate. For testing purposes, a pneumatic actuation system was used. However, it is anticipated that with the use of a hydraulic actuation system, much more positive actuation could be achieved. Combined with an appropriate control system, the actuation time could then be selected at will. After actuation of the reversing mechanism, the turbine continued to rotate for on the order of one to two seconds. It is anticipated that with a propeller installed, the reversing jets would interfere with the propeller inflow and reduce this time somewhat.

The numerical results of the testing and performance are presented in Figure 65 and Table 15. The thrust of the reversing mechanism is computed with the following formula:

$$T_R = \dot{m}\ (U_R \cos \alpha_R + Va).$$

Assuming the ship to be maneuvering from rest, then Va = 0.

$$T_R = (\rho\ Q)\left(\frac{Q}{A_R}\right) \cos \alpha_R$$

$$T_R = \frac{\rho\ Q^2 \cos \alpha_R}{A_R}$$

where:

T_R - reversing thrust

$\dot{m}$ - mass flow (sl/sec)

U_R - reversing jet velocity (ft/sec)

α_R - reversing jet angle - 37 degrees

ρ - fluid density - 1.937 sl/ft^3

Q - volume flow (ft^3/sec)

A_R - reversing nozzle area - 0.0213 ft^2

V_a - advance velocity

Also presented in Figure 65 is a curve of the ratio, R, of the reverse thrust to the equivalent propeller thrust as a function of pump speed. The equivalent propeller thrust is the thrust that would be developed by a propeller attached to the turbine, assuming the transmission to be operating at maximum efficiency. The assumed open-water propeller efficiency is 70%. The ratio R is calculated by noting first that at a given advance velocity, the equation for power is as follows:

$$H_{P_R} = T_R \, Va/550$$

$$H_{P_P} = T_P \, Va/550$$

where:

H_{P_R} - power developed by reversing jets

H_{P_P} - thrust horsepower of propeller

T_P - propeller thrust.

So:

$$R = \frac{T_R}{T_P} = \frac{H_{P_R}}{H_{P_P}}$$

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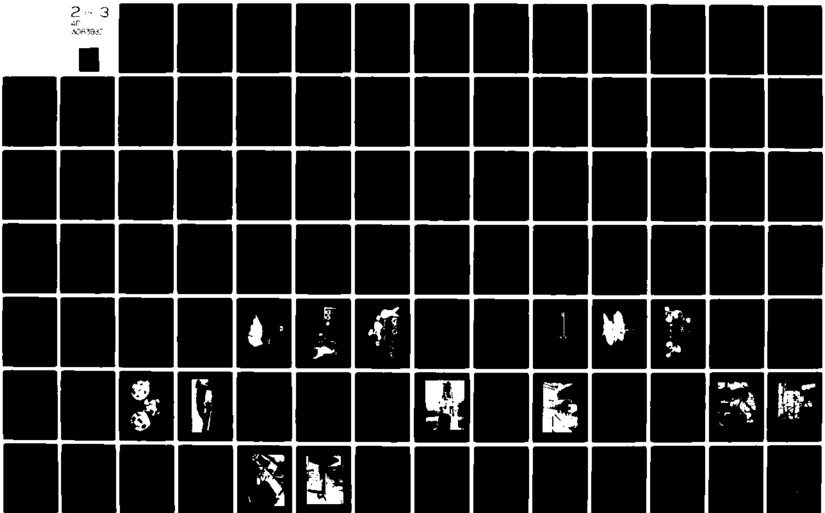

$$R = \frac{H_{P_R}}{0.7\ H_{P_T}}$$

where H_{P_T} is the maximum turbine horsepower developed at the appropriate pump speed. The reversing horsepower H_{P_R} is:

$$H_{P_R} = \frac{Q\ \gamma\ h}{550} \cos \alpha_R$$

$$H_{P_R} = \frac{Q\ \gamma\ (\frac{U_R^2}{2g})}{550} \cos \alpha_R$$

$$H_{P_R} = \frac{\rho\ Q^3}{1100 A_R^2} \cos \alpha_R$$

The reversing horsepower is assumed to remain constant with advance velocity, a conservative assumption.

The ratio R attains values between 40% and 48%, which means that at its best, the reversing mechanism can produce roughly half of the thrust in the reverse mode that a propeller would produce in the forward mode. The reversing thrust attains a maximum value of 29.2 lbf at a pump speed of 4000 RPM. Heavy cavitation was experienced during the test at 4000 RPM, so higher speeds were not tested. The data indicate, however, that at the design pump speed of 7500 RPM, the reversing mechanism

would produce 102 lbf of reversing thrust.

During the design of the reversing mechanism, several compromises were made which tended to reduce the performance of the mechanism. First, to ensure the structural integrity of the exit guide vane support without the need for lengthy and expensive detailed stress analysis, the reverse thrust nozzles were made deliberately small at 5/8 inch diameter for ten holes. With this arrangement, the pump flow was severely restricted and the pump operated at a flow coefficient between $\phi_p = 0.138$ to 0.158. Figure 66 shows that this is a very poor point of operation for the pump. The use of more holes, perhaps in two staggered rows, or of larger holes would increase the pump flow and the reversing thrust. It is estimated that the use of 10-3/4 inch holes or 15-5/8 inch holes would allow the pump to operate at approximately $\phi_p = 0.22$ which Figure 66 shows to be a much more advantageous point of pump operation. For simplicity, the nozzles were straight-walled. The use of a diffuser in the jet would yield more efficient reverse thrust production by reducing the jet velocity while keeping the mass flow constant. Finally, the installation of the unit in the lockout chamber, Figure 67, forced the reversing jet to turn through 153° in close proximity to the lockout chamber wall in order for the fluid to flow into the flow loop. In open water, this turning would not occur, and as such, the losses with this installation

are higher than would be experienced under actual operating conditions. The net result of the above observations is that, with careful design, the performance of the reversing unit could be greatly improved. It is anticipated that values of the ratio R on the order of 70% can be attained. It is worthwhile to note that this value compares with the value of reverse thrust obtained by reversing the direction of rotation of a fixed-pitch propeller. The reverse thrust of a fixed-pitch propeller, when reversing the direction of rotation, would be much less since the propeller efficiency would be well below the assumed 70%. Thus, the reversing mechanism, as tested, compares favorably with a fixed-pitch propeller operated in reverse. As a result of the tests, a reversing system can be developed with the new propulsion system for ships giving much shorter reaction time than any existing system and simultaneously developing large amounts of thrust. Thus, greatly improved ship maneuverability can be achieved.

The entire propulsion system had envisioned the installation of the hydraulic transmission model into a twenty-three inch diameter propeller. The turbine has a blade tip diameter of 5.7 inches and a preliminary analysis indicated that the turbine cylinder outer diameter and the propeller hub diameter would be seven inches. The seven inch hub diameter and the 23 inch propeller diameter result in a hub to tip ratio of

approximately 0.30. This is a desirable hub to tip ratio and has been used previously in the analysis of the new propulsion system.

For a transmission pump and turbine with the pump operating at its design point cavitation free operation to 5000 RPM can be expected on the basis of the transmission test. With a relatively small forward speed of 10 knots or above, the cavitation free operation of the transmission pump can be increased to 7500 RPM and the power input in the model will be on the order of 15 HP. These were the original design conditions of the transmission model.

If the transmission is installed in a 23-inch propeller and it achieves a transmission ratio on the order of 8 to 1, the propeller's speed will be 940 RPM with a transmission pump speed of 7500 RPM. For a ship application in the 15-HP range, the propulsion system competes with the usual outboard motor and propeller. For such outboard motor applications, the engine has a speed on the order of 5000 RPM with a propeller speed on the order of 3300 RPM. In these applications, the propeller diameter is on the order of 9 inches. Thus, by comparison the new propulsion system has a propeller which is approximately 2.5 times larger than the conventional propeller and the propeller speed is approximately 3.5 times smaller than the conventional propeller speed. It is assumed that for the above low power applications and the quoted speeds, the conventional system will have a propeller efficiency of 55% and a gear transmission efficiency of

90%. Thus, 7.4 HP will be transmitted as thrust in the water. Based on the transmission tests, a properly matched hydraulic transmission will have an efficiency of 80% and at the proposed propeller speed the propeller efficiency will be 75%. The ratio of propeller efficiency is indicated in Figure 17 for the above mentioned diameter and speed ratios. The absolute value of propeller efficiency is slightly lower in the above example than shown in Figure 17 as the low power units operate at a much lower Reynolds number. Thus, approximately 9.0 HP will be transmitted into the water. In addition, the larger propeller and lower shaft speed will be much less subject to cavitation which may result in additional efficiency increase. As a result in this specific application, the new propulsion system will show an efficiency improvement of approximately 22% over the conventional system. This increase in efficiency is the result of a lower speed propeller operating at higher efficiency. The increase in efficiency will result in an appropriate reduction in fuel consumption and an increase in range for the same fuel capacity. It is noted that the larger propeller will require a slightly deeper draft. The above example is a clear indication and an example that the new propulsion system can be used to improve overall system efficiency even though the efficiency of the hydraulic transmission is lower than that of the mechanical gear transmission.

VI. PROPULSION SYSTEM PERFORMANCE

A. Comparison of Efficiencies With Conventional Propeller

The comparison of efficiency with conventional propellers has been briefly discussed in Chapter III-D. The propeller driven by the new hydraulic transmission can have a higher efficiency than the gear driven propeller in a certain operating range due to the water discharge from the turbine in the hub of the propeller giving additional thrust. The new propulsion system has been studied covering a range of velocities of propeller to turbine discharge (Sheets, 1978-2). It is noted that for large values of turbine discharge thrust to total thrust together with high transmission efficiencies, the efficiency of the new propulsion system can easily exceed the efficiency of the gear driven propeller. The results indicate, as shown in Figures 12 to 15 that there exists an optimum turbine discharge to total thrust ratio which is a function of a variety of operating conditions. It is also affected by the ratio of turbine diameter to propeller diameter. These studies have been made assuming that the propeller diameter and the propeller shaft speed is the same for the standard gear driven propeller and for the new propulsion system. The transmission efficiencies are assumed to be 85% and 90%. These are conservative values for hydraulic transmission above 5000 HP, in view of the test data shown in Chapter V. The propeller efficiencies of 65%, 70%, and 76% are realistic as indicated in the literature mentioned in Chapter II.

It is well known that propeller efficiency increases with propeller diameter and that increased propeller diameter requires a slower shaft speed. In many gear transmissions, it is customary to have a two step reduction gear or otherwise limit the speed reduction to certain values so that the gear will not become too large. For high values of speed reduction, such as 40:1 and larger, the gear can become excessively large, heavy, and costly. In the case of the new hydraulic transmission, there is considerably more liberty in selecting the speed step down ratio. A larger step down ratio may require more turbine stages and perhaps a larger turbine diameter, but these features are generally not very costly and will not increase the weight of the transmission by a substantial amount. Figures 11 and 17 present the calculated data of propeller efficiency versus propeller shaft speed ratio. They indicate that it is very beneficial for the new propulsion system to drive a propeller of relatively large diameter and make use of the increase in propulsive efficiency permitted by such a design. The propulsive efficiency can be increased by over 10% if the propeller diameter is appropriately increased and the shaft speed reduced. The new propulsion system can easily increase the total propulsive efficiency by providing a hydraulic transmission with a higher transmission ratio driving a slower rotating propeller of a larger diameter.

B. Contra-Rotating Efficiency

The use of contra-rotating propellers with the new propulsion system is comparatively easy because the prime mover can drive a

single rotation pump which in turn will drive contra-rotating turbines, each carrying an individual propeller. Thus, the ship's hull is penetrated with a high speed shaft requiring only a simple relatively small shaft seal. This is a substantial simplification compared to previous mechanically driven contra-rotating propellers which require large diameter, slow speed concentric shafts with associated slow speed shaft bearings. In addition, each of the shafts requires an individual relatively large shaft seal resulting in considerable mechanical complexity. The performance of contra-rotating propellers has been analyzed in Comstock (1967) indicating an efficiency increase of approximately 9% over single rotation propellers. The data are summarized in Table 2 giving efficiency increase, diameter decrease, and weight increase. It is noted that the diameter decrease will result in a more favorable ratio of turbine discharge thrust to propeller thrust.

It should be noted that with the new hydraulic transmission, very slow shaft speeds and high values of a step down ratio are easier to accomplish with a contra-rotating turbine as compared to a single rotation turbine. This fact is well known to the designers of turbines. As a result, the new transmission system offers a real opportunity for very high propulsive efficiencies with contra-rotating propellers of slow shaft speed. Figure 17 shows the efficiency increase which is possible with slow speed contra-rotating propellers. Additional studies of contra-rotating propulsion systems are recommended in the light of the substantial gains in efficiency which are possible with such systems.

C. Weight Savings of the New Propulsion System

The new propulsion system eliminates the reduction gear and, as a result, there is a considerable saving in weight. Figure 68 shows the weight of reduction gears as a function of horsepower and shaft speed. The data are taken from Harrington (1971). In practice, gears have been built with higher values of weight per horsepower as well as smaller values of weight per horsepower. Specifically, planetary gears usually weigh less. Planetary gears are more costly than conventional gears in the lower horsepower range, 4000 through 12,000 horsepower with single reduction. But they are less costly than conventional gears in the higher horsepower range, 30,000 through 45,000 horsepower with double reduction. Figure 68 indicates that decreased shaft speeds result in a substantial increase in gear weight. Thus, it is an important consideration that lower shaft speeds, which can result in improved propulsion efficiency, require reduction gears of considerably larger weight and cost. The new propulsion system, while eliminating the reduction gear, replaces a good performer which is reliable and requires little maintenance cost. The new propulsion system must be engineered from the beginning to have maximum reliability and low maintenance costs.

It is noted that for a conventional gear driven propeller, there is a slow speed shaft from the reduction gear to the propeller. This shaft not only has a large diameter, but it is frequently of considerable length, and thus of very substantial weight. These large diameter long shafts also require shaft couplings and shaft bearings of considerable size and weight. In

addition, there is a shaft seal of considerable size where the propeller shaft penetrates the ship's hull. Thus, a slow speed propeller requires a substantial weight associated with the propeller shaft, bearings, couplings, and shaft seal. There are no detailed data in the literature regarding the weights of these components similar to Figure 68 for reduction gears. However, it is obvious that for the slower shaft speeds, these weights increase by a substantial amount.

The new propulsion system has a high speed shaft of small diameter and associated small diameter bearings, shaft couplings, and shaft seals. It is also possible to reduce the length of the shaft because the power turbine is usually of smaller height than the reduction gear and, therefore, can be moved farther aft in the ship.

In a preliminary weight estimate, comparing a conventional propeller drive with the new propulsion system, it is assumed that the propeller diameter for both systems is identical. With this assumption, it is estimated that the propeller weight of the conventional propeller is about equal to the weight of the propeller of the new propulsion system. In the new propulsion system, the hub of the propeller is hollow and, in this hollow hub is incorporated the stationary and rotating turbine blades and the associated supporting shaft. Thus, the new propulsion system includes in the propeller weight the hydraulic transmission turbine structure. The basic concept is shown in Figure 5 indicating that even the incorporation of the transmission turbine into the hub of the propeller still leaves a partially hollow hub filled with water as

compared to the solid metal structure of the conventional propeller.

The single stage pump with associated pump housing, guide vanes, reversing mechanism, and transition structure from pump to turbine are all external to the ship's hull in the new propulsion system. Thus, it can be assumed that their collective weight is less than the weight of propeller shaft, seal, and bearings of the conventional propeller. There are additional savings in weight of the new propulsion system over the conventional propeller since there are no foundations in the ship required for the support of the reduction gear. Depending on the type of ship and the location of the gear, these savings in weight can be substantial. Omission of the conventional reversing mechanism will produce additional savings in weight.

It is noted that the total savings in weight will be larger for the higher shaft reduction values. These are usually associated with higher propulsive efficiencies. The above preliminary studies indicate that the new propulsion system will save a total amount of weight equal about 1.5 of the gear weight as indicated in Table 16. This includes, in addition to the weight savings of the gear itself, savings in weight for the foundations, shafts, seals, and bearings. These savings in weight can be used for potential improvements to have a slower shaft speed, associated with higher propulsive efficiency, or it can be used to have contra-rotating propellers with the new propulsion system, or a combination of both as indicated in Table 17. The savings in weight can also be used for an appropriate increase in payload or

the entire ship can be built smaller and lighter for improved performance due to reduction in required power and associated reduction in fuel consumption. For certain ships, where the reduction gear is supported by special sound or shock mounts, the weight savings may be considerably larger.

D. Cost Estimate

A preliminary cost estimate of the hydraulic transmission has been made and for this cost study it is assumed that the entire ship cost remains unchanged except for the cost of the transmission. That assumption is based on using the savings in weight, discussed above, entirely for increasing the payload. The increased revenue from this increased payload is not considered in this study which evaluates acquisition costs only.

As a first step, the costs of the reduction gear are estimated. Preliminary information from gear manufacturers shows gear cost to be on the order of \$10/lb for the large horsepower sizes or 50,000 horsepower, increasing to about \$15/lb for a 10,000 horsepower gear. These costs are for commercial gears and are price quotations received in 1978. For military gears, there is a slightly higher cost having a range of \$12/lb for a 50,000 horsepower gear, increasing to \$20/lb for a 10,000 horsepower gear. These costs are associated with conventional turbine drives where usually two turbines are connected to a reduction gear driving a single shaft and propeller. The data are presented in Table 18 and Figure 69, and they have been transformed into gear cost as a

function of shaft horsepower and shaft speed for both commercial and military gears shown in Table 19. The gear costs are graphically summarized in Figure 70 indicating gear cost in dollars as a function of horsepower. In Table 19, an average gear reduction ratio has been assumed be ng in the range of 20:1 to 30:1. In this preliminary analysis, the cost savings due to omitting foundations, reversing mechanisms, and similar savings in weight, as discussed above, have not been considered.

In order to estimate the cost of the hydraulic transmission, it has been assumed that none of the development costs should be directly charged to the hydraulic transmission. Since no transmission of this type has been built, the preliminary cost estimate has given consideration to present costs for gas turbines when ordered in quantities of one. The hydraulic transmission does not require the fuel system, combustion chamber, and other components requiring special consideration due to the high temperature of a gas turbine. Thus, it is felt reasonable to deduct from the cost of a gas turbine such items as combustion chamber, fuel nozzles, fuel control, fuel treatment system, as well as parts of the hot section, including flexible joints, special exhaust ducts, long transmission shafts, and compensation devices for axial and radial expansion for the gas turbine structure due to heat. With these assumptions, it is estimated that the cost of the hydraulic transmission will equal about 60% of the gas turbine costs for the same power. It is noted that since the hydraulic transmission uses water, an incompressible fluid, the turbine stages and their

nozzles can be of identical configuration in all stages. This results in substantial manufacturing cost reduction over existing gas turbines. On the basis of the above assumptions, the cost data of Table 20 have been prepared on the basis of 1978 costs. The hydraulic transmission cost data are shown in Figure 70. Since the costs of the hydraulic transmission in Figure 70 does not include the cost savings for foundations, shafts, etc., as mentioned above, it is believed to be realistic and conservative within the information presently available.

VII. PROPULSION APPLICATIONS AND SYSTEM IMPROVEMENTS

A. Applications for the New Propulsion System

The applications for the new propulsion systems fall into two groups. The first group relates to those ships and propulsion applications where the new propulsion system will show distinct advantages over existing gear driven or electric propulsion systems. The second group relates to systems with new characteristics for which presently no comparable propulsion systems exist.

The new propulsion system saves weight as well as space compared to existing systems, and if appropriately designed, can have substantially higher propulsive efficiency. These advantages will be most significant for ships of relatively high power and high speed. Into this group fall a number of military vessels as well as fast container ships. In addition, special high speed vessels belong into this group, such as hydrofoils and surface effect ships.

Another application relates to ships which use full power only a relatively small percentage of the time. In these cases, gears are designed for maximum power output. Into this group of ships belong a number of naval vessels, Coast Guard cutters, and some commercial ships.

The new propulsion system can generate a relatively larger thrust from a fixed propeller diameter because it generates thrust

over the entire propeller diameter due to the thrust from the turbine discharge. Conventional propulsion systems generate no thrust over the approximately inner 20% of the propeller diameter because the hub of the propeller is a core of metal with no fluid discharge. Thus, the new propulsion system generates more thrust from a fixed propeller diameter. This is an advantage for large tug boats which have a limited propeller diameter due to draft limitations. Into this group belong river tug boats and ships having similar applications.

The new propulsion system is particularly applicable to ships where the transmission and propeller is located an appropriate distance below the water surface. In these cases, cavitation can be substantially reduced or avoided. Into this group belong ships with large drafts, submersibles, as well as oil drilling rigs where the propulsion system is located at a depth substantially below the water surface. In addition, special vehicles which have relatively deeper draft should be considered such as the SWATH design ships.

Contra-rotating propellers have been used on an experimental basis in a very limited number of ship applications due to the complexity of the mechanical drive system. The new propulsion system permits the application of contra-rotating propellers in a much simpler way by using contra-rotating water turbines. The contra-rotating propulsion system will result in some additional cost and weight over the single rotation propeller system. However, the improved propulsive efficiency will result in increased

speed, or the saving of fuel, or a combination of both. Contra-rotating propellers have applications for a large number of vessels where fuel costs are important. Such applications will become more numerous and economically more important as the cost of fuel increases.

The new propulsion system will permit a variable transmission ratio by using a transmission pump with adjustable blades. Thus, the transmission can provide an entire range of speed reductions, and the entire propulsion system can be optimized as a function of load requirements. The variable transmission ratio may also be of particular advantage to ships with dual prime movers, such as diesel and gas turbines, or generally for two types of prime movers, for low speed and high speed. The new propulsion system with an adjustable blade pump can also vary the ratio of propeller thrust to turbine discharge thrust, and it can vary propeller speed according to a predetermined function. These are all new characteristics which may be of importance in general naval applications.

B. Contra Rotation

Of all the applications discussed in Chapter VI, the most significant application of the new propulsion system is the use of contra rotation in both the propeller and the transmission turbine because it gives an inherent increase in efficiency. If the contra rotation is coupled also with a decrease in propeller shaft speed, the increase in propulsion efficiency can easily reach 30%

or more. This is a most important increase in propulsive efficiency which will result in substantially lower fuel and operating costs. Thus, the new propulsion system offers an opportunity in substantial fuel savings which, in turn, could permit smaller fuel tanks for the same range of operation. This, in turn, could result in a smaller ship of higher speed and a substantial increase in overall performance. The total increase in efficiency has the following components for a vessel with speeds on the order of 25 knots if all efficiency improvements are used to reduce the size of the ship. The efficiency increase amounts to the following values: contra rotation 9%, reduction in propeller shaft speed 10%, reduction in ship size due to omission of gear weight and space 3.5%, reduction in ship size due to fuel savings as the result of the first three items of efficiency increase 6.5%; improvements in operating efficiency due to the variable transmission ratio 2.5%. Thus, a total efficiency improvement of 31.5% is possible for the above case and thus, will also amount to a substantial reduction of ship acquisition and operating costs. It is recognized that the use of contra rotation with the new propulsion system will require a substantial development for both the propeller and the transmission. In addition, the incorporation of a contra-rotating propulsion unit may also require research and modification of the entire ship's configuration. However, such changes will be beneficial to ship drag and ship performance because with a contra-rotating propulsion unit the ship operates in a symmetric flow medium.

C. Turbine Hub to Tip Ratio

The turbine hub to tip ratio has been selected conservatively in conjunction with the experimental model, and thus, a fairly high hub to tip turbine diameter ratio exists in the model transmission turbine. This, in turn, requires a carefully designed exit diffuser, and this exit diffuser requires a predetermined length to have the proper diffusion downstream of the turbine. In the existing model, this diffuser is not optimum due to space limitations. The existing test data suggest that turbines with a lower hub tip ratio may be beneficial in reaching increased overall efficiency. They would also require a diffuser considerably smaller in size and simultaneously having a higher efficiency because less energy transformation is required downstream of the turbine exit. Since the turbine exerts a certain amount of flow resistance on the pump discharge flow, it should be possible to incorporate a higher degree of diffusion into the transition section between pump and turbine. It is also noted that in larger and higher powered units, the length between pump discharge and turbine entrance will be extended over a longer distance, and this, in turn, will permit more diffusion between pump and turbine. The present findings suggest that considerable research should be done in this flow area between pump discharge and turbine entrance. When this research is done, it may also be possible to structurally combine the pump discharge vanes and the turbine entrance nozzle. If this is possible, then the flow between pump discharge and turbine entrance may have a spiral configuration, thus permitting a substantially larger flow deceleration in the same axial

length between pump discharge and turbine entrance. This will permit turbines of much lower hub to tip ratio. Overall, this will lead to a shorter turbine exit diffuser, a shorter transition piece between pump and turbine, and a simplified structure. The reduction in overall length will also reduce the overall length of flow in the hydraulic transmission, and it should, in turn, increase overall efficiency.

D. Pump Specific Speed

The pump specific speed selected for the model transmission is relatively high. However, the pump specific speed is relatively low when analyzing the requirements for hydraulic transmissions of high speed reduction ratios. For a speed reduction of 30:1 or 40:1, a pump of higher specific speed should be used. The tests with the present small model suggest that pumps of these higher specific speeds can be built with high efficiency, particularly in the larger power sizes. The extremely good efficiencies achieved with the existing small model pump also suggest that the pump can be built with a relatively smaller diameter resulting in improved cavitation performance. The existing tests indicate that the critical element of the transmission, namely the pump, can be designed to meet the necessary requirements for the transmission and with additional research, improved efficiency and cavitation performance will be possible.

E. Efficiency

The present studies have analyzed the new propulsion system essentially as a replacement for existing ship propulsion considering ship speed in the conventional range of 15 to 30 knots. Under these conditions, the ratio of turbine tip to propeller tip ratio is on the order of 0.3. For special high speed applications such as hydrofoils and higher speed vessels, the ratio of turbine diameter to propeller diameter can be substantially increased. This, in turn, will also increase the ratio of turbine discharge thrust to propeller thrust. For these conditions, the new propulsion system will have inherently higher efficiencies than the conventional propeller system. If the turbine discharge thrust is further increased, the new propulsion system may to some degree, compete with water jet propulsion. However, since it still uses a propeller, the total mass flow will be larger than that of a water jet propulsion, and thus, the new propulsion system will have a higher efficiency for those applications where it can be incorporated in a ship design.

The present tests have shown a remarkably good efficiency for both the pump and the turbine. However, additional efficiency improvements are quite likely to be achieved, as discussed in the preceding sections of this report. Thus, with additional research, the efficiency of the hydraulic transmission can be increased. The assumed 90% transmission efficiency is quite conservative for large units and values of higher transmission efficiency may be possible in the future.

F. Larger Unit

The dimensions of the present prototype unit were selected to be compatible with the test facilities at the University of Rhode Island. The speed reduction ratio was kept moderate at a value of 10:1 to 7:1 in order to reduce the total number of turbine stages. As a result, the maximum speed of the pump is 7500 RPM, and the propeller maximum speed is 1000 RPM. The propeller diameter of 23 inches has been limited by the configuration of the towing tank. The turbine diameter of 5.7 inches has been selected to be on the order of 0.25 of the propeller diameter. The pump diameter of four inches is smaller than the turbine diameter and is controlled by the axial velocity through the pump. The entire design contains many compromises, and it is noted that the propeller speed is very low when compared to existing practice for small boat propellers. The propeller speed for outboard motors is approximately half the engine speed and is on the order of 3000 RPM.

The basic dimensions of the transmission are such that the same design can be used for higher horsepower ratings by increasing the speed of the prime mover and the propeller. These possibilities are shown in Table 21. In this table, number 1 is the model which has been tested. Units 2 to 7 have the same physical size but are capable of transmitting more power with increased prime mover speed. In the table, a transmission efficiency of 75% has been used for all data which is quite realistic and is based on the test data for the model. This efficiency is probably low for

the higher power units 5, 6, and 7 due to the change of efficiency with Reynolds number. The table shows prime mover speeds up to 30,000 RPM resulting in propeller speeds of 3000 RPM. As an example, a transmission for about 500 horsepower would require a prime mover speed of about 24,000 RPM with a propeller speed of approximately 2400 RPM. These are values which could be considered for special ships such as hydrofoils or SES types. Such a unit will not be physically larger than the present unit. The above values are mentioned to have an indication of changes in transmission size with an increase in power. For a transmission of substantially more power, higher strength materials may be required, and a careful stress analysis is needed due to the larger forces involved. In addition, a careful cavitation analysis is required particularly for the pump impeller. Higher shaft speed pumps will require the 50% reaction pump for improved cavitation performance. Cavitation may limit the maximum allowable tip speed of the transmission pump impeller. It may also be desirable to change the transmission ratio of the unit to reduce propeller speed and change the thrust ratio between propeller and turbine exhaust for the higher power transmission in order to provide for optimum propulsive efficiency. The above considerations indicate that a 500 horsepower unit would not be very much larger than the existing prototype.

Since the proof of the concept of the new propulsion system demonstrated successful operation of the entire unit as well as good efficiencies, the next step in this program should be the

consideration of a larger unit. The previously discussed increase of power of the existing small unit is limited to very high speed vessels due to the effects of cavitation. For more conventional speeds, a unit of approximately 50% to 100% increase in dimensions will provide for a substantial increase in power. For such a unit, a power range on the order of 200 to 1000 horsepower can be achieved, and such a unit would be applicable for a range of existing ships. The next unit should be designed for a higher speed reduction ratio on the order of 20:1 or 30:1. This would involve double the number of turbine stages. With such a unit, the full range of capability of the new propulsion system should be explored. In order to cover the potential of this propulsion system and assure the maximum increase in efficiency, it is suggested that the unit be built with contra rotation. This would involve considerably more research and development, and it also may require adjustable inlet nozzles into the first turbine stages to permit full control of turbine and propeller speed. Such a unit would demonstrate the high efficiency and small size of this propulsion system, and it would permit an assessment of the performance of the system, including such features as variable transmission ratios and total weight and cost of the unit.

G. <u>Ship Control</u>

A new concept of ship directional control is possible with the new propulsion system, since the hydraulic transmission is of the axial flow type and is located outside the ship's hull. The

water discharge from the transmission turbine can be used for ship control by deflecting it in a predetermined direction. This is accomplished by means of a set of movable hydrofoils installed in the transmission turbine discharge duct. By deflecting the turbine discharge water in the appropriate direction, the action of the rudder and/or control surfaces can be supplemented or control surfaces can be eliminated. This control system can provide a wide range of thrust forces and directions. By adjusting the pump vanes, the ratio of flow energy going through the transmission and, subsequently, to the movable hydrofoils can be increased compared to the energy transmitted to the propeller. Thus, very large amounts of power are available for ship control, if needed. This system is shown in principle in Figure 71.

The proposed control system in its simplest form will provide flow vectors in a single plane and thus will replace the rudder of the ship. However, there will be significant changes compared to the normal rudder action. More specifically, very large forces of directive thrust will be available at low and zero ship speed. At high speed, the angle and amount of the flow thrust vector can be adjusted to meet requirements. In the above described version, the control system consists of a simple actuator controlling the discharge angle of the vanes.

A second version of the control system is also shown in Figure 71, and it can provide the same flow directivity of the vanes, but in addition, it will provide for the rotation of the entire vane system, thus giving a two axis flow control and associated thrust force vector. Such a system will permit both steering and

diving control for submersibles. On surface ships, this system will permit corrections in the ballast system of the ship or to correct the ship to an even keel in case of uneven load distribution.

The new concept of ship control provides for large directive thrusts at very low speeds, thus greatly improving maneuverability in harbors and in docking operations. For special ships such as hydrofoil or surface effect ships, the proposed control system could provide bow thruster action by simply connecting the forward water jet thruster through a pipe and control valve to the high pressure section of the hydraulic transmission. This is shown in Figure 72.

The proposed control system can be designed to be substantially more efficient than the standard rudder. This is possible by operating the propulsion system in such a mode that, at steady state conditions requiring no ship control action, the water flow through the turbine discharge section is at low velocity and low rate of flow, since the thrust from the turbine discharge provides only a small percentage of the total thrust. The majority of the thrust comes from the propeller driven by the hydraulic transmission. For maximum ship control, however, the adjustable vanes in the transmission pump can be controlled so that a much larger flow will flow through the transmission. The thrust from the propeller becomes then relatively smaller, and the thrust from the transmission turbine discharge becomes larger. Thus, high water velocity exists in the ship control section only when control is required.

In case ship control is not required, the water velocity is low and so is the corresponding drag or power loss due to the ship control section.

The above system is only described in basic principles, and additional studies are needed to optimize this system for a specific application. For this purpose, a ship should be selected so that all aspects of this ship control system can be properly studied and evaluated. The briefly described ship control system is presented to indicate that the new propulsion system for ships offers many new capabilities.

H. The Ship and Propulsion System

The new propulsion system uses a pump as part of the hydraulic transmission. For single propeller ship applications, the pump is usually arranged at the aft end of the ship and symmetric to the ship centerline. In such a location, the water flow into the pump will ingest a substantial part of the ship boundary layer. Since the pump is absorbing the full power of the prime mover, substantial quantity of water and boundary layer are removed from the ship's hull. The designer can control the amount of water the pump will deliver as the pump can use large quantity of water flow and a small head or smaller quantity of water flow and a larger head within certain limits. The designer can also select the configuration upstream of the pump and select a design which is favorable to ship boundary layer removal and water flow entrance into the pump.

By considering boundary layer removal from the ship's hull, entirely new ship configurations are possible and will permit the design of lower drag ship hull forms. The boundary layer removal will permit steeper pressure gradients in the aft end of the ship and it can result in a substantial extension of the laminar flow region in the forward part of the ship. Essentially such new ship configurations will have a larger beam and shorter length for the same tonnage or ship volume. Thus, these ship forms have a smaller surface area under otherwise similar conditions and the smaller surface area will result in lower ship drag. The new ship configurations have the potential of substantial reduction in ship drag and thus will improve overall ship performance. In addition, the transmission turbine discharge flow will inherently accelerate the boundary layer near the roots section of the propeller blades and improve propeller efficiency and performance.

The new ship configurations can be considered with the previously described ship control system. At the end of this five-year study of the new propulsion system for ships it becomes evident that the system's total capabilities go much further than just ship propulsion. The optimum application will be a totally new ship system and propulsion configuration which will lead to the largest total efficiency increases. Some qualitative thoughts have been given to these potential improvements in ship performance. The improvements can easily amount to a total efficiency increase having minimum values of 30%, when only considering conservative effects of contra rotation, savings in weight and space

together with a more efficient propeller shaft speed, to maximum values well above 100%, when considering optimizing all systems including new ship configurations for hull shape and ship control. This subject deserves considerable effort and study in the next phase of the program in order to quantify the improvements to the total ship system.

VIII. CONCLUSIONS

A. Proof of Concept

The research work done under this contract covers the final phase of research on an entirely new concept for ship propulsion. In the slightly over five years since its inauguration, the University of Rhode Island has sought to develop a better understanding of the complex relationships involving ship propulsion, ship control and a host of independent problems related to hydrodynamics, structural mechanics, efficiency, weight and cost. The studies and tests have brought proof that the proposed concept is simple and capable of high efficiencies. The simplicity of structure was demonstrated in the first air model which also showed a remarkably low number of individual parts when constructing the entire unit. The second air model provided a demonstration of using contra-rotating propellers in connection with the new propulsion system. It demonstrated the simplicity of contra-rotating propellers when being driven by the contra-rotating hydraulic turbines of the transmission.

A small hydraulic transmission water model was built and tested, showing excellent efficiencies. A 90% efficiency was demonstrated in the test of the very high specific speed pump and the five-stage axial water turbine achieved 91% efficiency during the tests. This high turbine efficiency was attained even though the turbine was of the inverted design, had a high turbine hub to tip ratio and had straight blades which had to be used to save time and

money. By designing the turbine with twisted blades and other improvements, a slightly improved total efficiency can be expected.

The total transmission efficiency showed values up to 71% which is excellent considering the small size of this transmission and the many technical innovations. The test data indicate that with a third pump, or a matched pump and turbine combination, a total transmission efficiency in excess of 80% could be expected with the small 4" pump and 5.7" turbine of the water model. The water model also demonstrated the simple structure and the relatively small number of parts which are required for this propulsion system. The entire mechanical design, the small thrust bearing and the main bearings of the transmission model worked quite satisfactorily. The water model also brought proof of two features which no gear transmission is capable of, namely the variable transmission ratio and a very fast acting reverse thrust system. Thus, the demonstration of the water model was a success in proving the concept of the hydraulic transmission and the new propulsion system.

The experimental data clearly indicated that a large transmission can be designed and built with efficiencies above 90%.

An analysis was made to compare the "New Propulsion System for Ships" with the standard gear transmission and the results indicate that the new propulsion system equals, and with design improvements, it can exceed the efficiency of the standard gear transmission. The improvement of the new propulsion system over

the standard gear transmission is the result of an increase in thrust due to the transmission turbine discharge flow. It became evident that the new propulsion system can generate a higher thrust for the same propeller diameter and it can produce a higher propeller efficiency for the same thrust. The new propulsion system has weight savings on the order of 1.5 times the reduction gear weight and there are equivalent savings in space. These savings can be used to improve ship's performance, increase pay load, or a combination of these factors.

A cost analysis has shown that the new propulsion system will be less expensive than a gear transmission. This is easily understood due to the substantial reduction in weight of the new propulsion system compared to the gear transmission. The boundary layer control effects of the new propulsion system on the ship's hull and the propeller have been assessed but not quantified, and they provide additional improvements in overall ship system efficiency.

B. Future Applications and the Total Ship System

The applications for the new propulsion system fall into two groups. The first group relates to those ships and propulsion applications where the new propulsion system will show distinct advantages over existing gear driven or electric propulsion systems. The second group relates to systems with new characteristics for which presently no comparable propulsion systems exist. The new propulsion system is capable of improved efficiency

because it can drive a slow speed contra-rotating propeller of high efficiency. Thus, it should find applications for ships where high efficiency and low fuel consumption are important. They include fast container ships, special vessels, hydrofoils, surface effect ships and applications where the propeller is deeply submerged.

It now appears that the new propulsion system is more promising than initially anticipated. That feeling was reinforced by outside interest in the work, by presentations given and papers offered at scientific and development conferences where people from different backgrounds and disciplines were brought together. The new propulsion system is generating thrust by both the propeller and the transmission turbine discharge while receiving its hydraulic power from the transmission pump. The system provides for two thrust generating systems which are concentrically arranged around the propeller shaft. Thus, the new propulsion system can provide thrust through the hydraulic transmission pump even when the propeller is damaged or otherwise malfunctioning.

And finally, at the end of the five years the new ship propulsion concept has been studied, analyzed, and a research model has been built and tested. The studies have indicated that the optimum performance of the new concept should include the entire ship system, namely ship configuration and ship control, in addition to ship propulsion. The transmission pump can be used for boundary layer control on the ship's hull and lead to entirely new and more efficient hull configurations. The transmission turbine

discharge flow can be vectored in the desired direction, thus eliminating the ship's rudder. The concept is entirely new, and as with any new undertaking, additional improvements can be made with research on the reversing system, the transmission, as well as the integration of the propulsion system into the ship.

However, the data indicate it is now beyond any question or doubt that the concept is sound and offers a great potential improvement in ship design, propulsion and control. Efficiency increases over existing systems can have minimum values of 30% when only considering contra rotation, weight and space savings, together with more efficient propellers, to values well over 100% when optimizing all systems including hull shape and ship control. Like any new concept its full realization is incremental, moving from one phase to another in a logical sequence.

The future program should include the development of a larger unit with a power range on the order of 200 horsepower to 1000 horsepower and provisions for a low cavitation pump. An installation in a ship should be contemplated so that the new propulsion system can be integrated with the ship's hull and eventually a ship control system may be added. For such an application, a test program can be envisioned to permit research of the new propulsion system in a ship's installation, and to provide experimental data regarding boundary layer effects on the propeller and the ship's hull.

A tentative conclusion to be drawn from the work done thus far is that it is entirely possible to advance ship design and

propulsion to a new level of performance through the implementation of a three point program extending over three or four years costing approximately one million dollar to 1) design and 2) build a scaled up model that will be 3) tested at sea. There is an abundance of research, analytical and experimental data to substantiate the validity of this recommendation as being the next logical step to take.

IX. ACKNOWLEDGMENTS

The Principal Investigator has greatly appreciated the financial support this project has received from the Office of Naval Research, Code 230, during the period 1974 to the present and from the Maritime Administration Marine Science Program during the last three years when the project was jointly supported by both agencies. Mr. Ben Friedman, ONR Code 230, and Mr. Frederick Seibold, Manager, Marine Sciences Program, Maritime Administration, have also given encouragement, made many technical suggestions, and substantial contributions to the overall program, all are most gratefully acknowledged.

Professor Tadeusz Kowalski has been Associate Investigator covering all aspects of naval architecture and he has been helpful in many ways on this project. Mr. Albert P. Davis, Jr. as Research Associate at URI and later as President of Offshore Subsea Systems has supervised all manufacturing, testing, and was in charge of designing and building the models. He has made extraordinary contributions to the entire project. A number of students have been supported by and worked on this project, and their contributions are acknowledged. The University of Rhode Island supported this project by manufacturing many parts in the Engineering Instrument Shop, and the University made available the facilities of the Department of Ocean Engineering for testing the propulsion unit in the laboratory at the Bay Campus. The Principal Investigator gratefully acknowledges the support and advice he

has received from the many individuals who assisted in the execution of this research project. The project covered the entire range of research design, building and testing the "New Propulsion System for Ships". Ms. Linda Wilkinson deserves special thanks for her patience, for typing, for assembling the many reports and for dedication and contributions to the entire project during the last five years.

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FIGURE 1

RELATIONSHIP OF SUBCAVITATING EFFICIENCY TO ROTATIVE COEFFICIENT

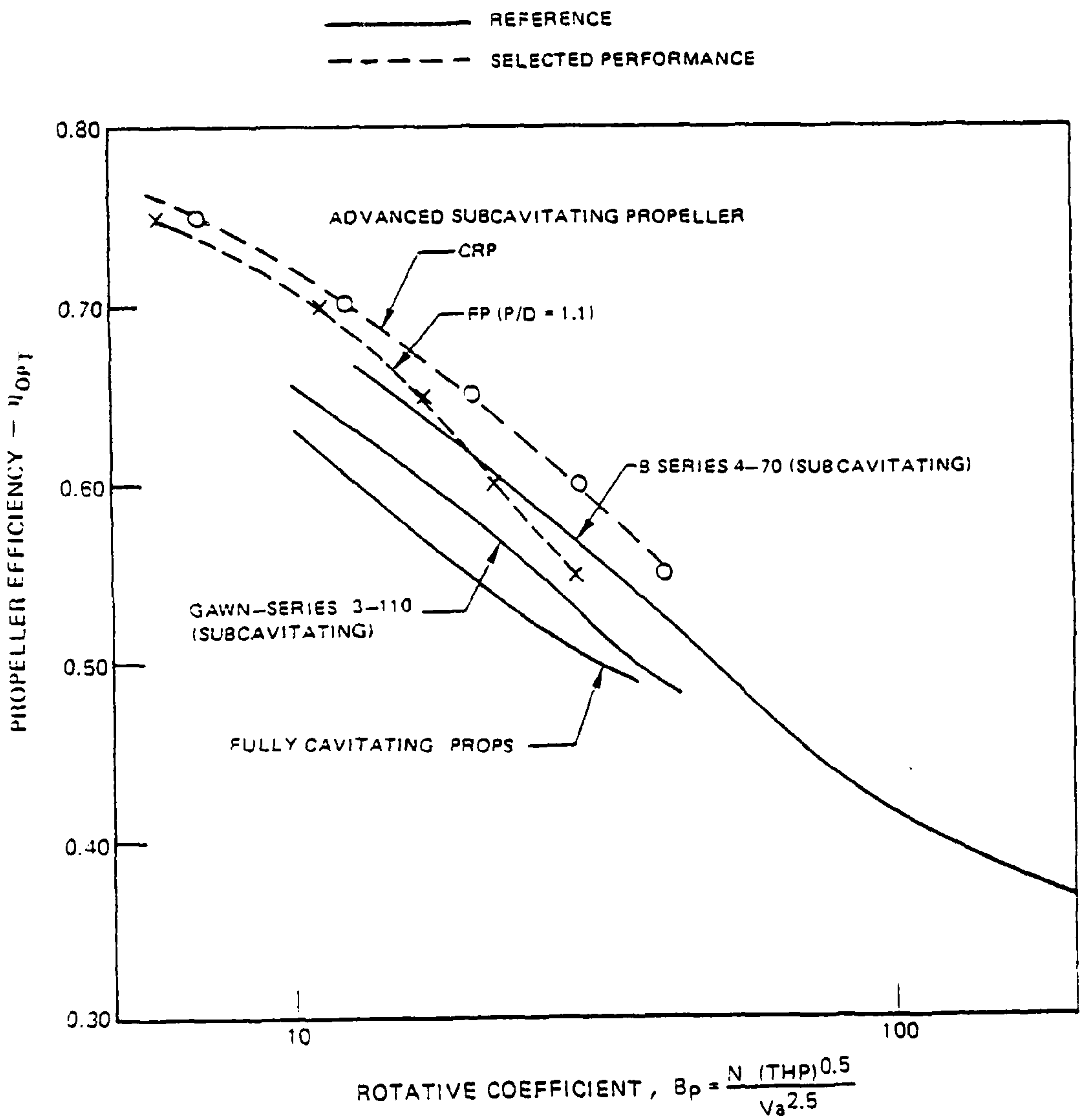


FIGURE 2

SUBCAVITATING PROPELLER DESIGN CHART FOR 26–KNOT AUXILIARY SHIPS

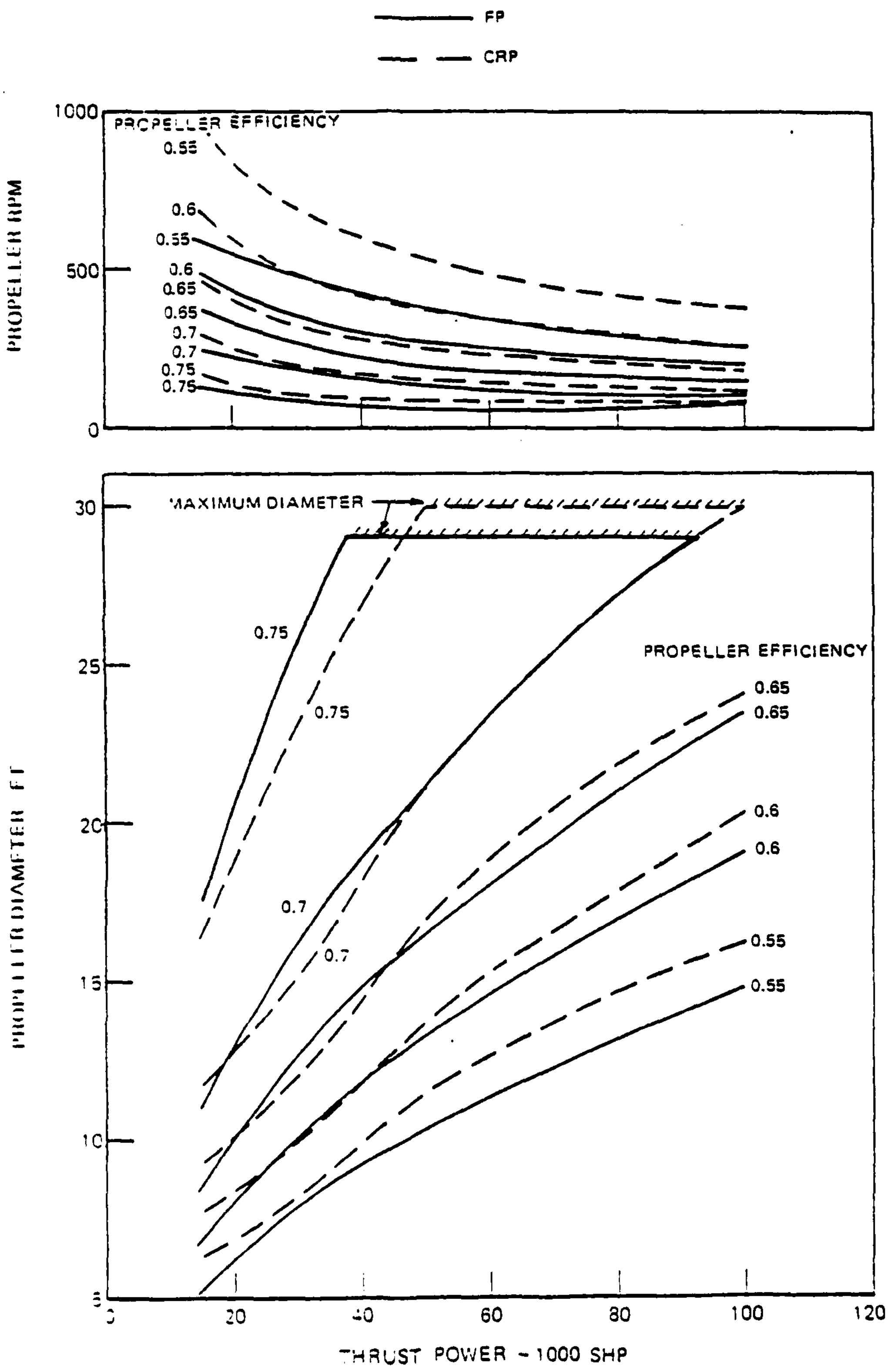


FIGURE 3

SUBCAVITATING PROPELLER DESIGN CHART
FOR 35–KNOT SHIPS

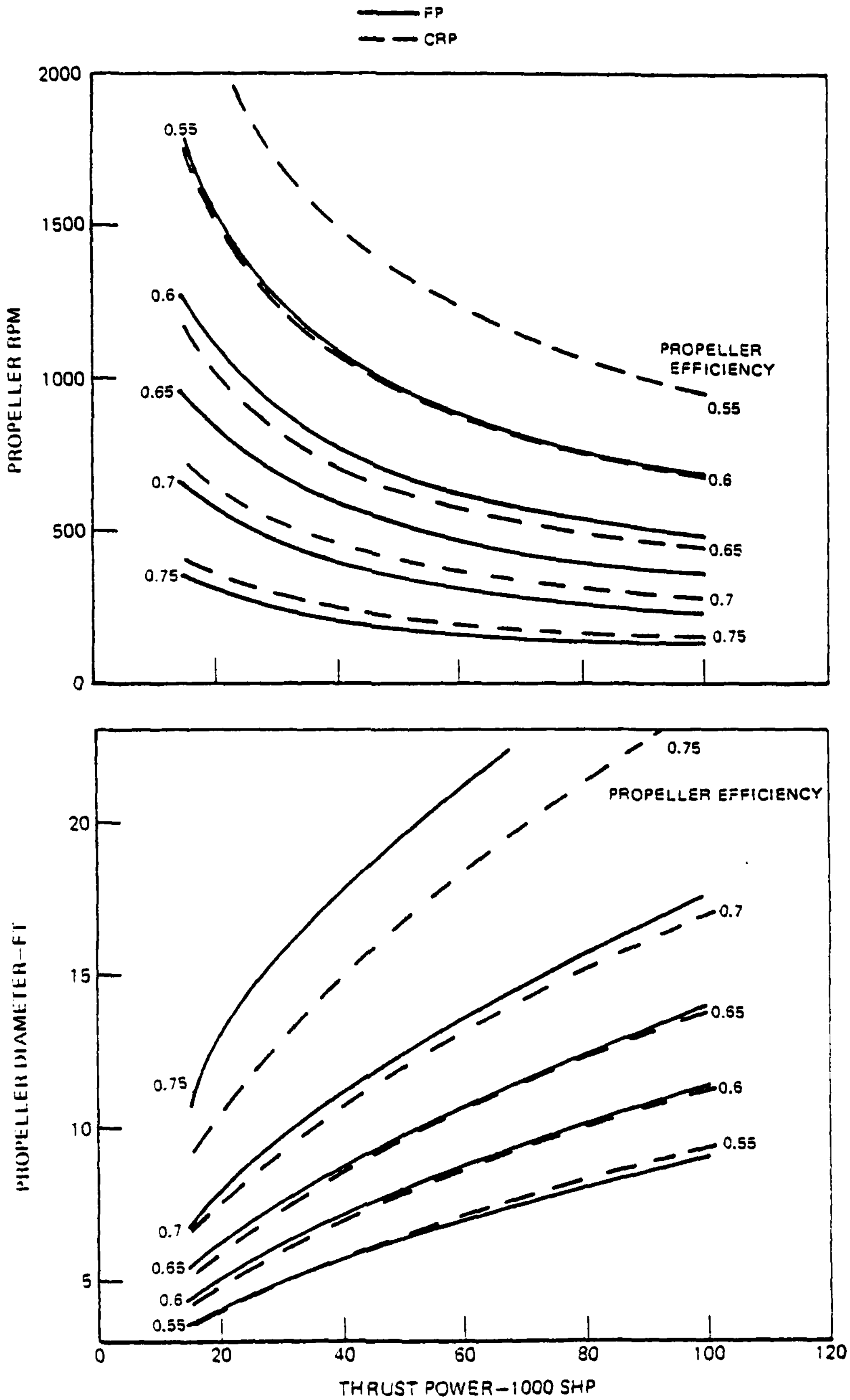


FIGURE 4

SUPERCAVITATING PROPELLER DESIGN CHARTS
FOR 50–KNOT SHIPS

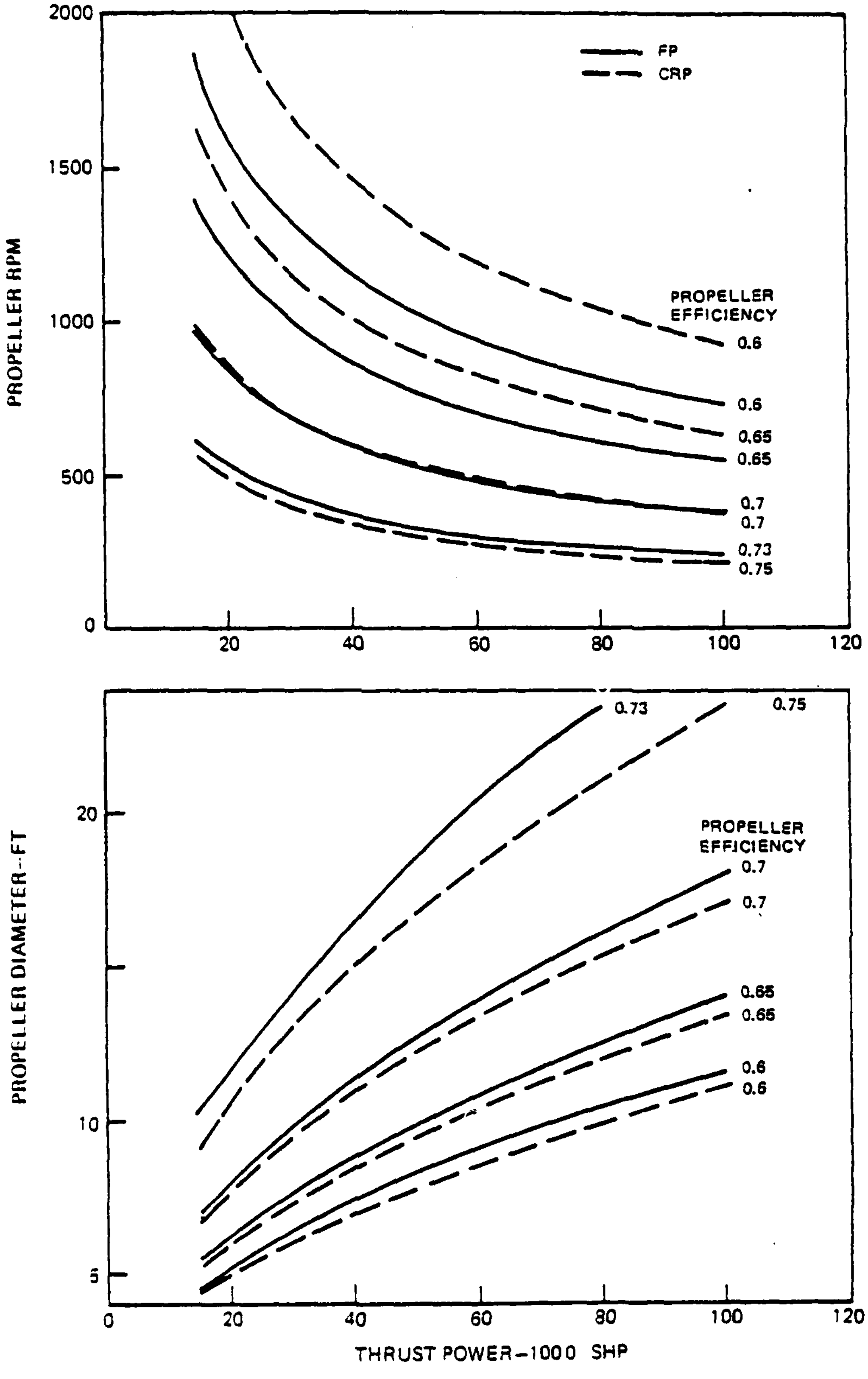
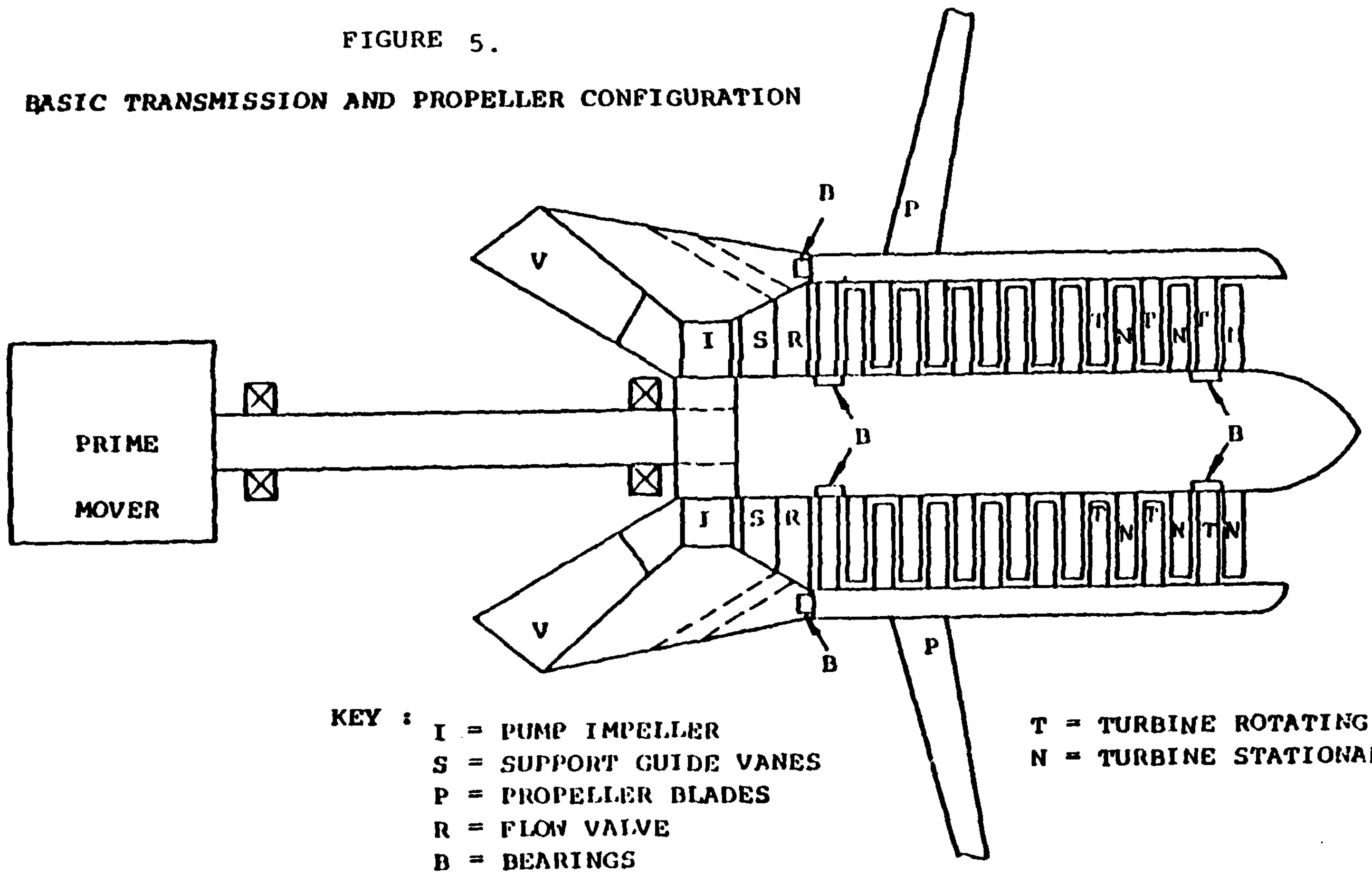


FIGURE 5.

BASIC TRANSMISSION AND PROPELLER CONFIGURATION



KEY :
I = PUMP IMPELLER
S = SUPPORT GUIDE VANES
P = PROPELLER BLADES
R = FLOW VALVE
B = BEARINGS
V = INLET VANES

T = TURBINE ROTATING VANES
N = TURBINE STATIONARY VANES

FIGURE 6.

PUMP EFFICIENCY vs SPECIFIC SPEED

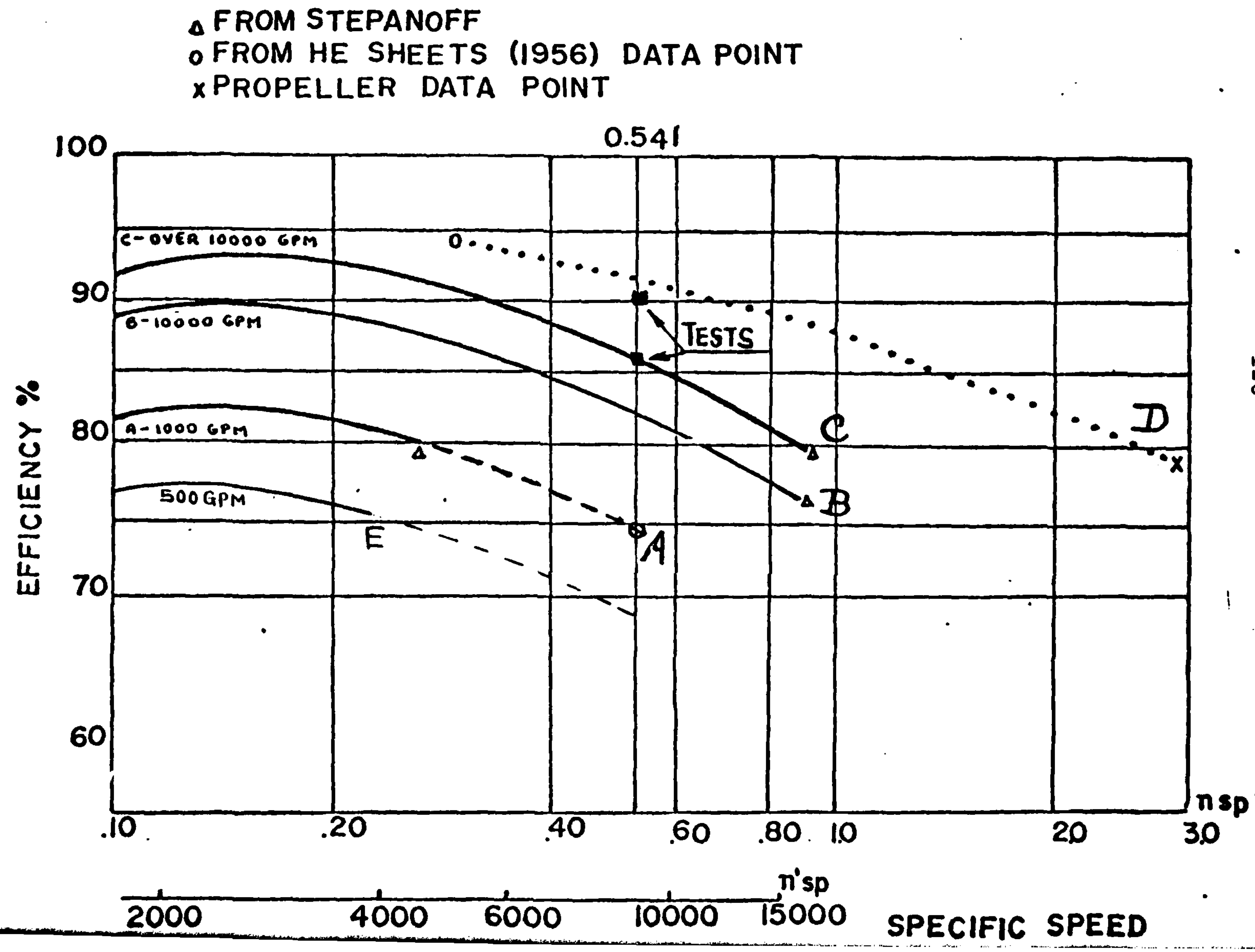
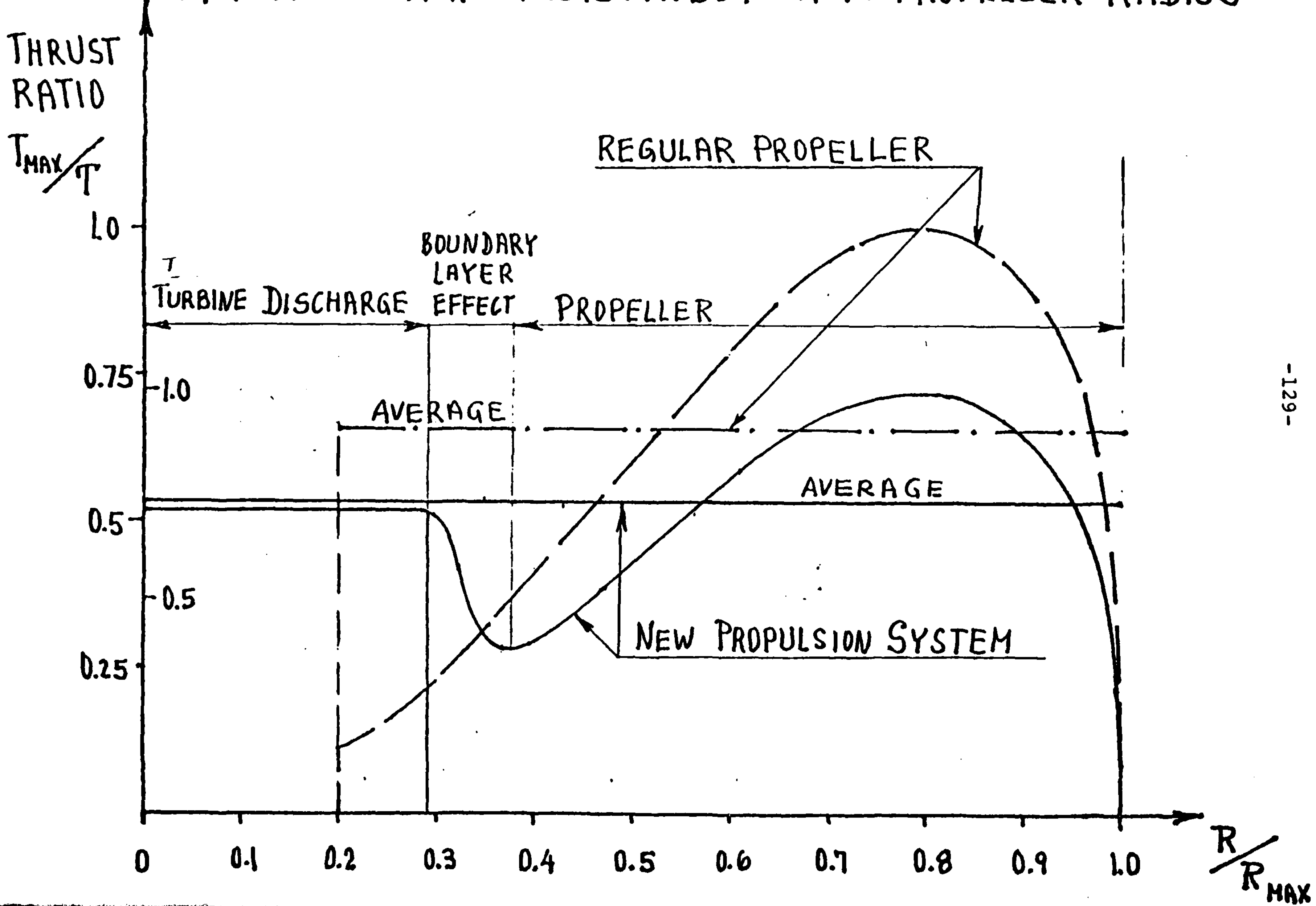
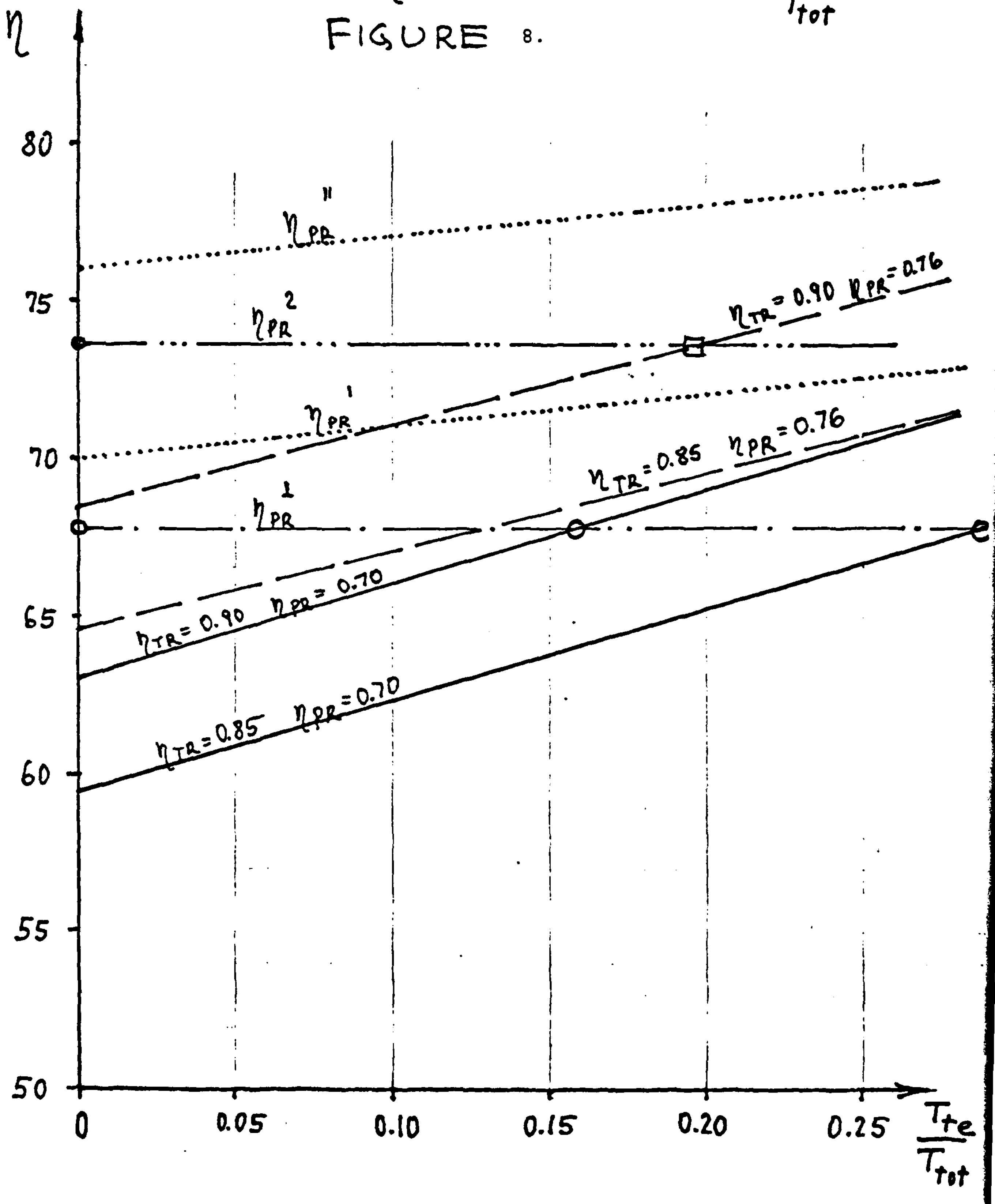


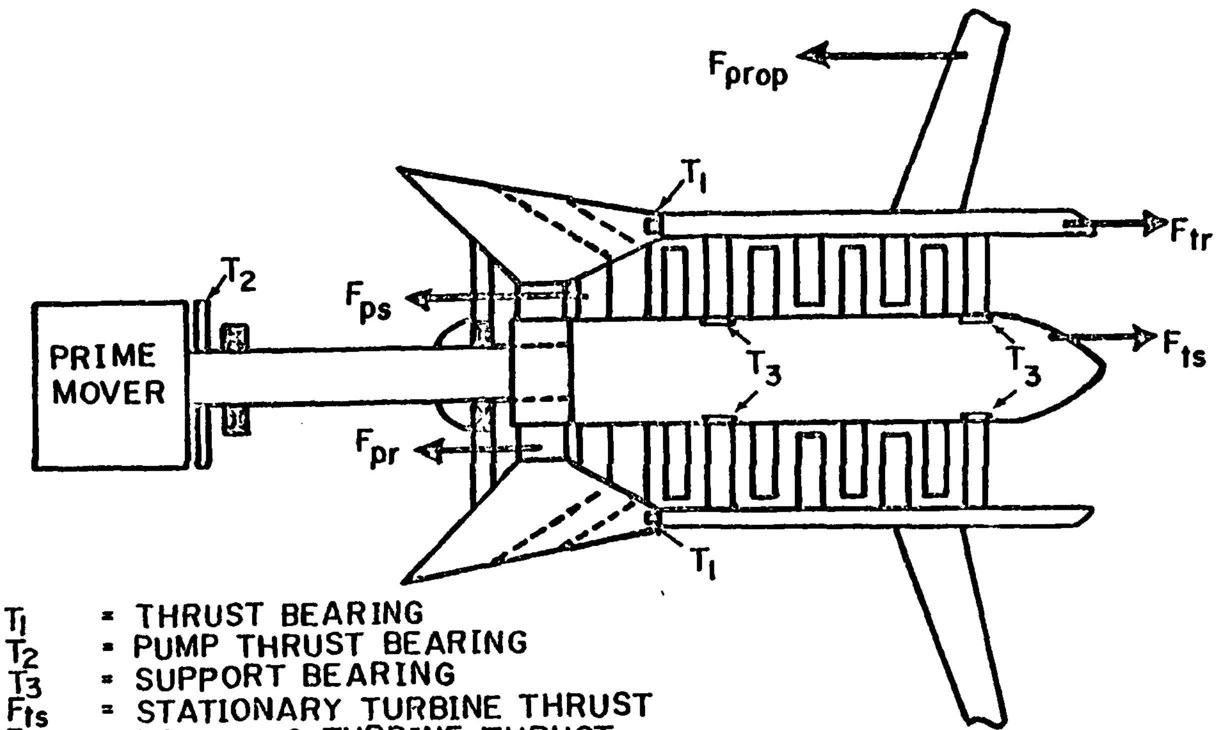
FIGURE 7. THRUST DISTRIBUTION VS PROPELLER RADIUS



EFFICIENCY η VS THRUST RATIO T_{te}/T_{tot}

FIGURE 8.



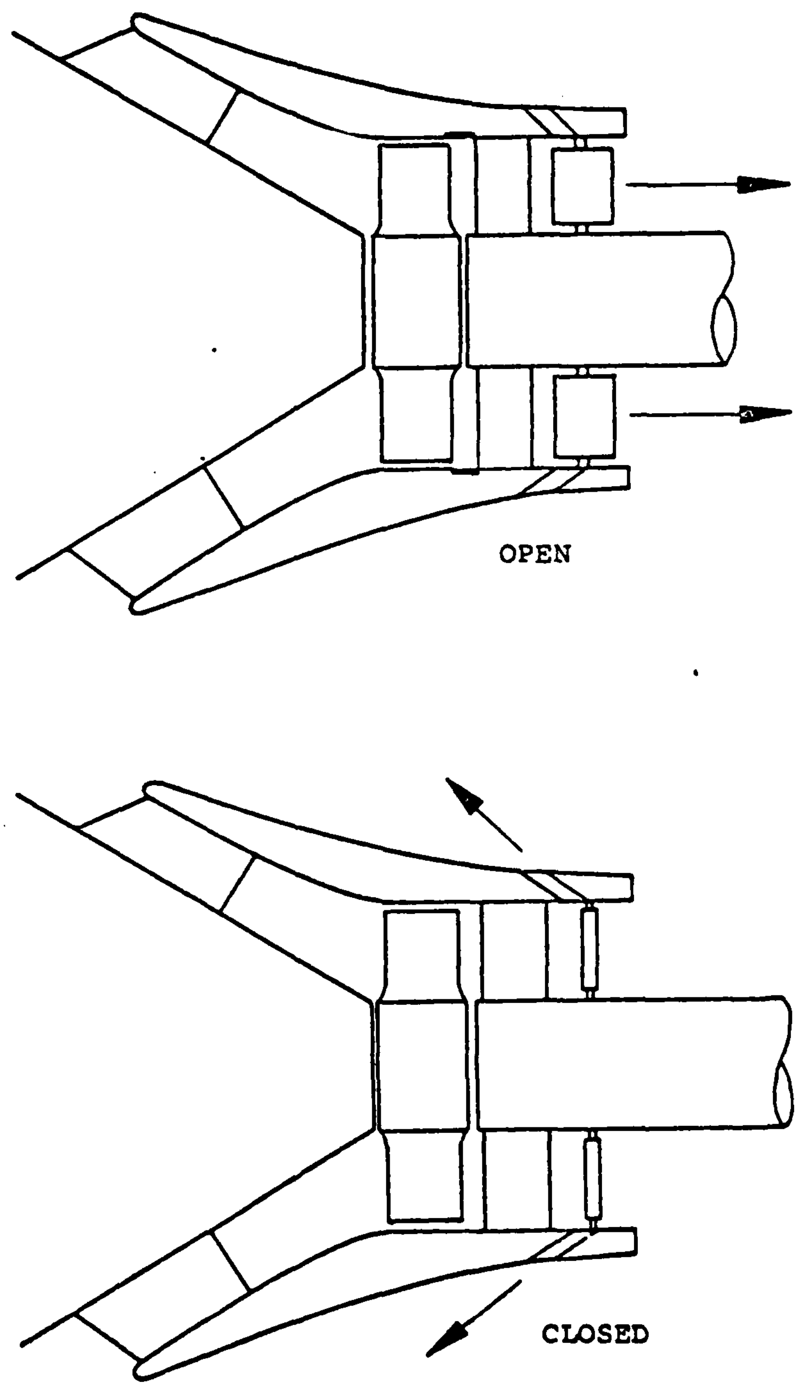


T_1 = THRUST BEARING
T_2 = PUMP THRUST BEARING
T_3 = SUPPORT BEARING
F_{ts} = STATIONARY TURBINE THRUST
F_{tr} = ROTATING TURBINE THRUST
F_{pr} = PUMP THRUST ROTATING
F_{ps} = PUMP THRUST STATIONARY
F_{prop} = PROPELLER THRUST

Fig. 9 Thrust Bearings and Thrust Forces

FIGURE 10.

REVERSING MECHANISM



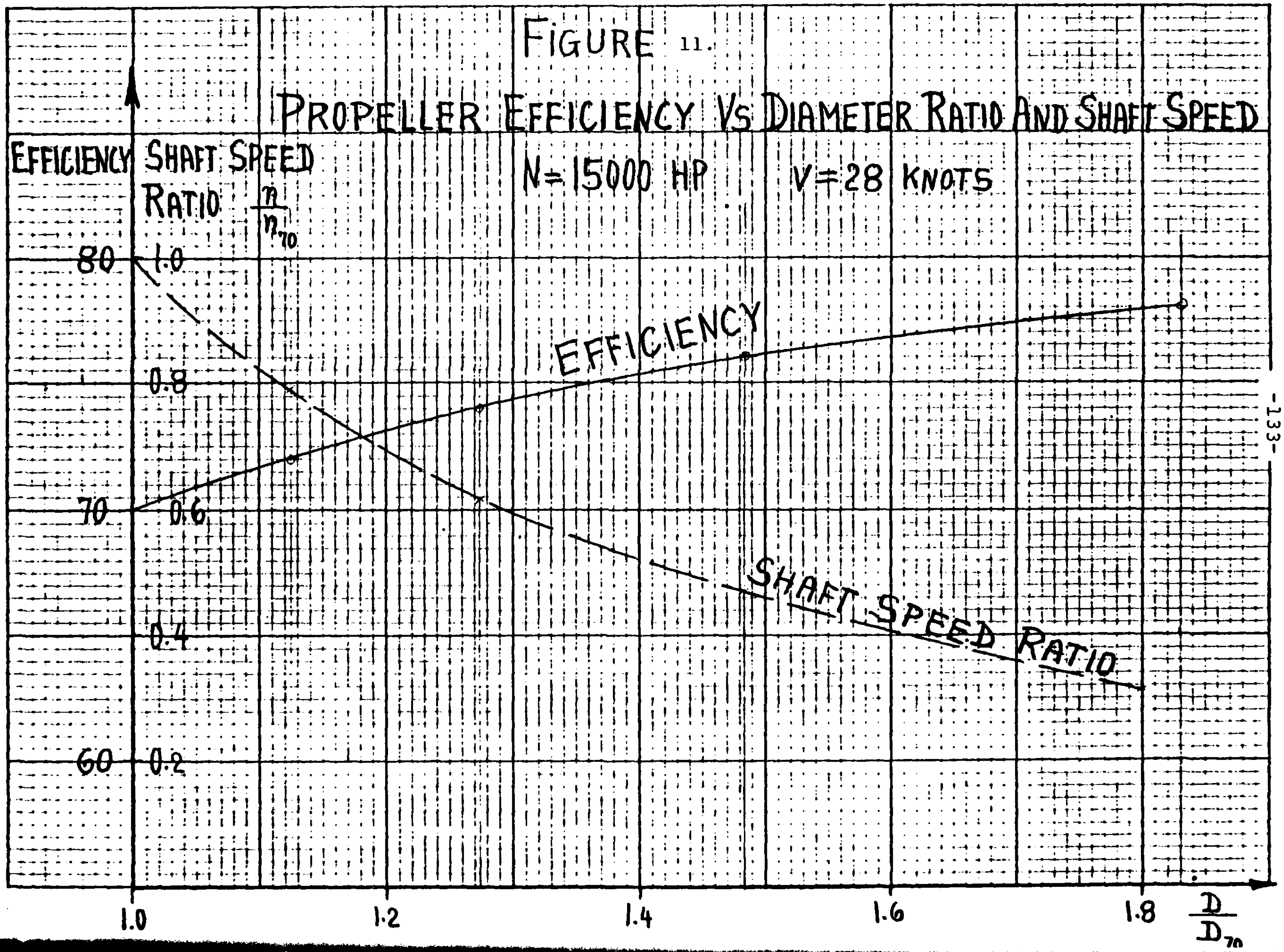
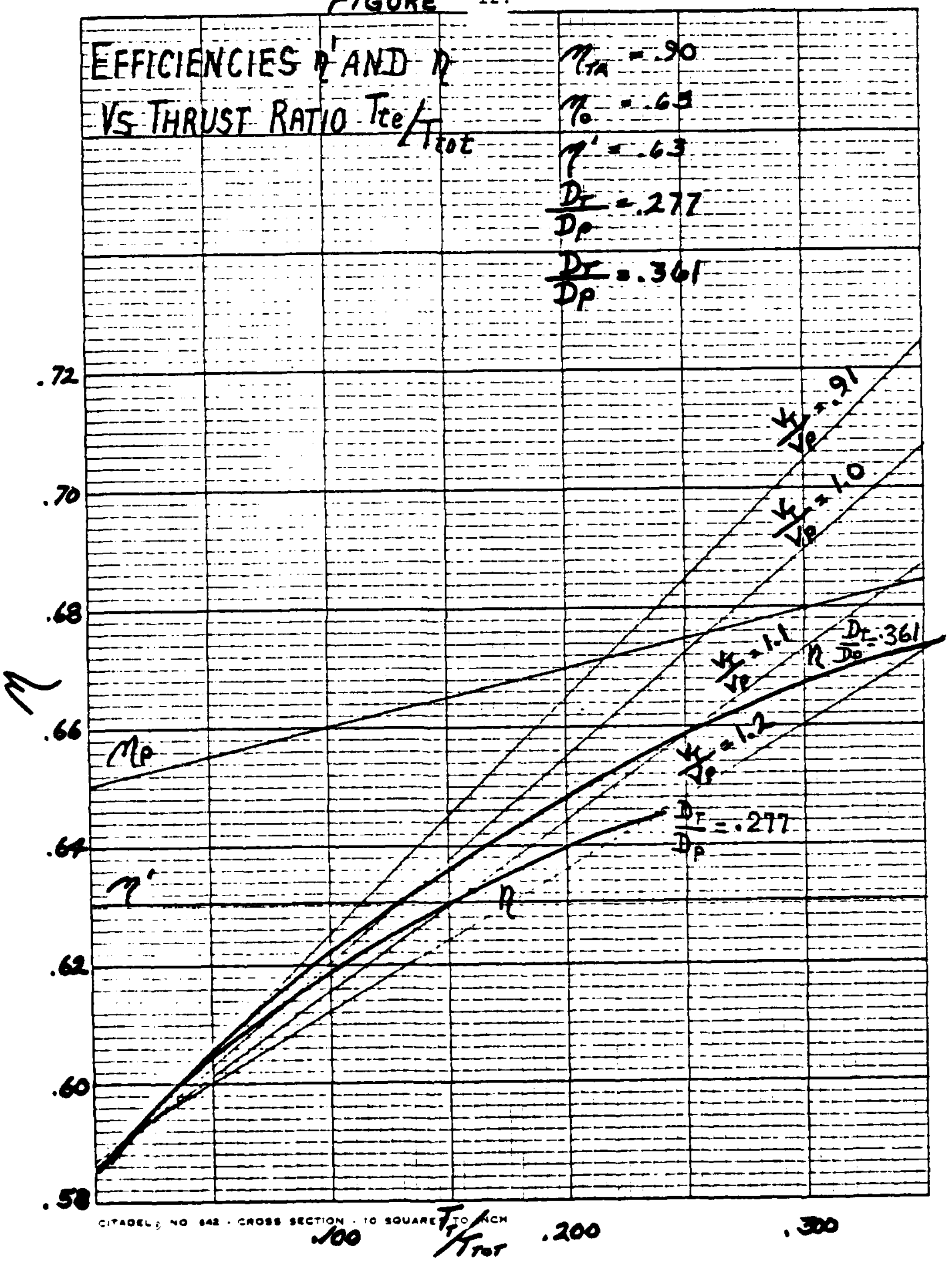


FIGURE 11.
PROPELLER EFFICIENCY Vs DIAMETER RATIO AND SHAFT SPEED
N = 15000 HP
V = 28 KNOTS
EFFICIENCY
SHAFT SPEED RATIO $\frac{n}{n_{70}}$
80
70
60
1.0
0.8
0.6
0.4
0.2
EFFICIENCY
SHAFT SPEED RATIO
1.0
1.2
1.4
1.6
1.8
$\frac{D}{D_{70}}$

FIGURE 12.


EFFICIENCIES η' AND η
VS THRUST RATIO Tte/Ttot
ηTR = .90
ηo = .63
η' = .63
DT/DP = .277
DT/DP = .361
VT/VP = .91
VT/VP = 1.0
VT/VP = 1.1
VT/VP = 1.2
DT/DP = .361
DT/DP = .277
ηP
η'
η
.72
.70
.68
.66
.64
.62
.60
.58
.100
.200
.300
TT/TTOT
CITADEL NO 642 · CROSS SECTION · 10 SQUARES TO INCH

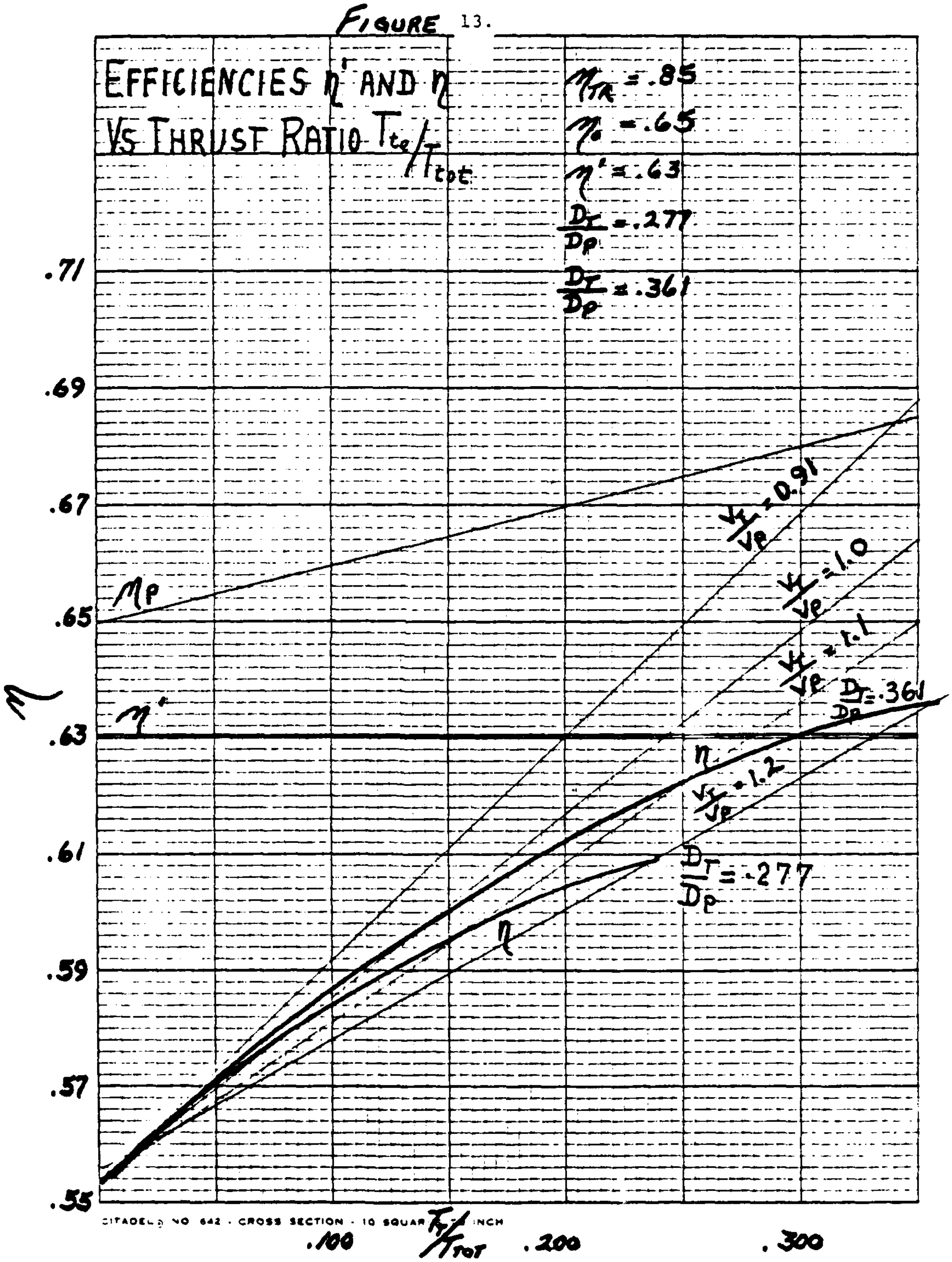


FIGURE 13.
EFFICIENCIES η' AND η
VS THRUST RATIO Tte/Ttot
ηTR = .85
ηo = .65
η' = .63
DT/DP = .277
DT/DP = .361
.71
.69
.67
.65
.63
.61
.59
.57
.55
ηP
η'
η
VT/VP = 0.91
VT/VP = 1.0
VT/VP = 1.1
VT/VP = 1.2
DT/DP = .361
DT/DP = .277
CITADEL NO 642 · CROSS SECTION · 10 SQUARES INCH
.100
.200
.300
TT/TTOT

FIGURE 14.

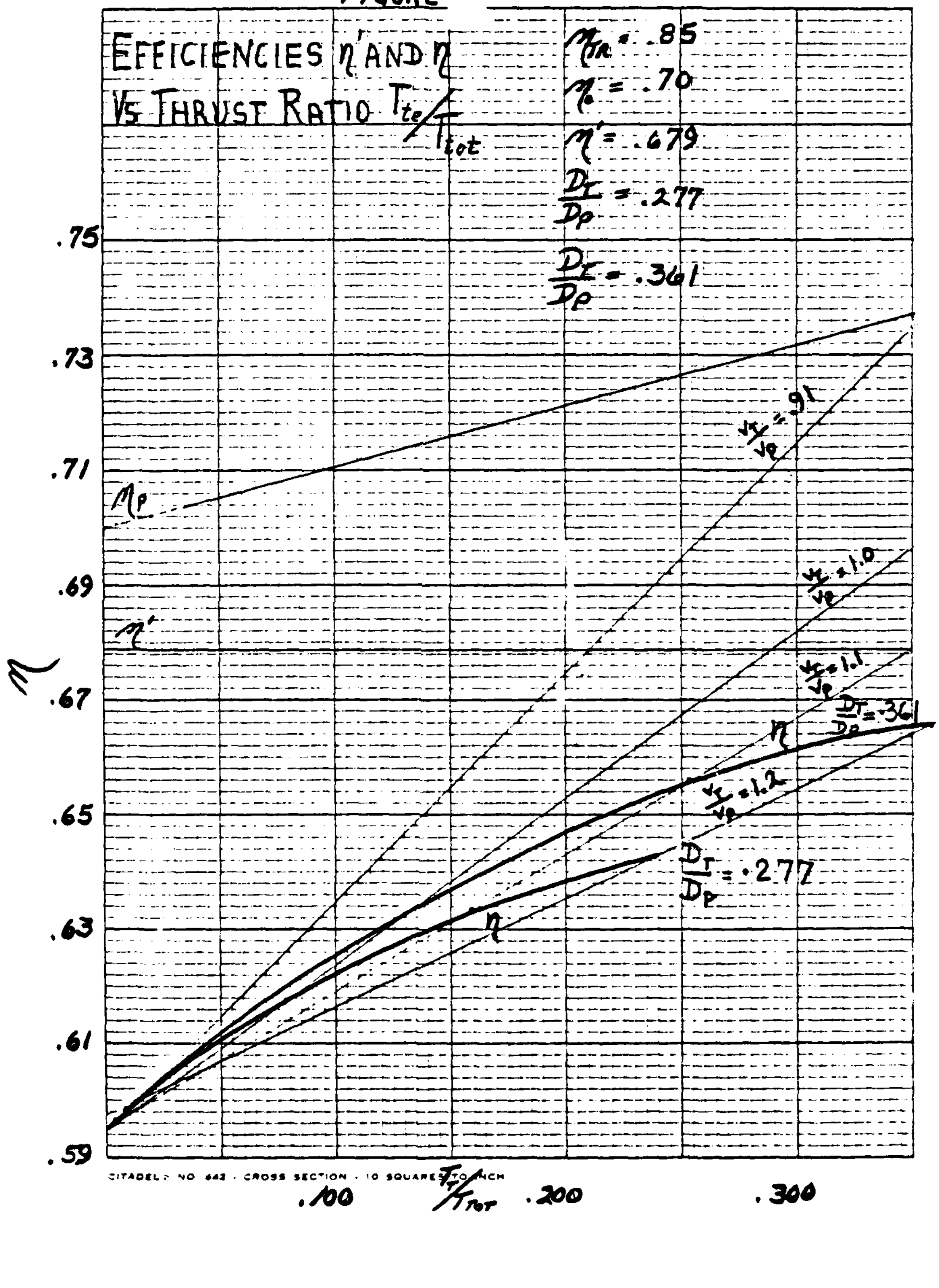
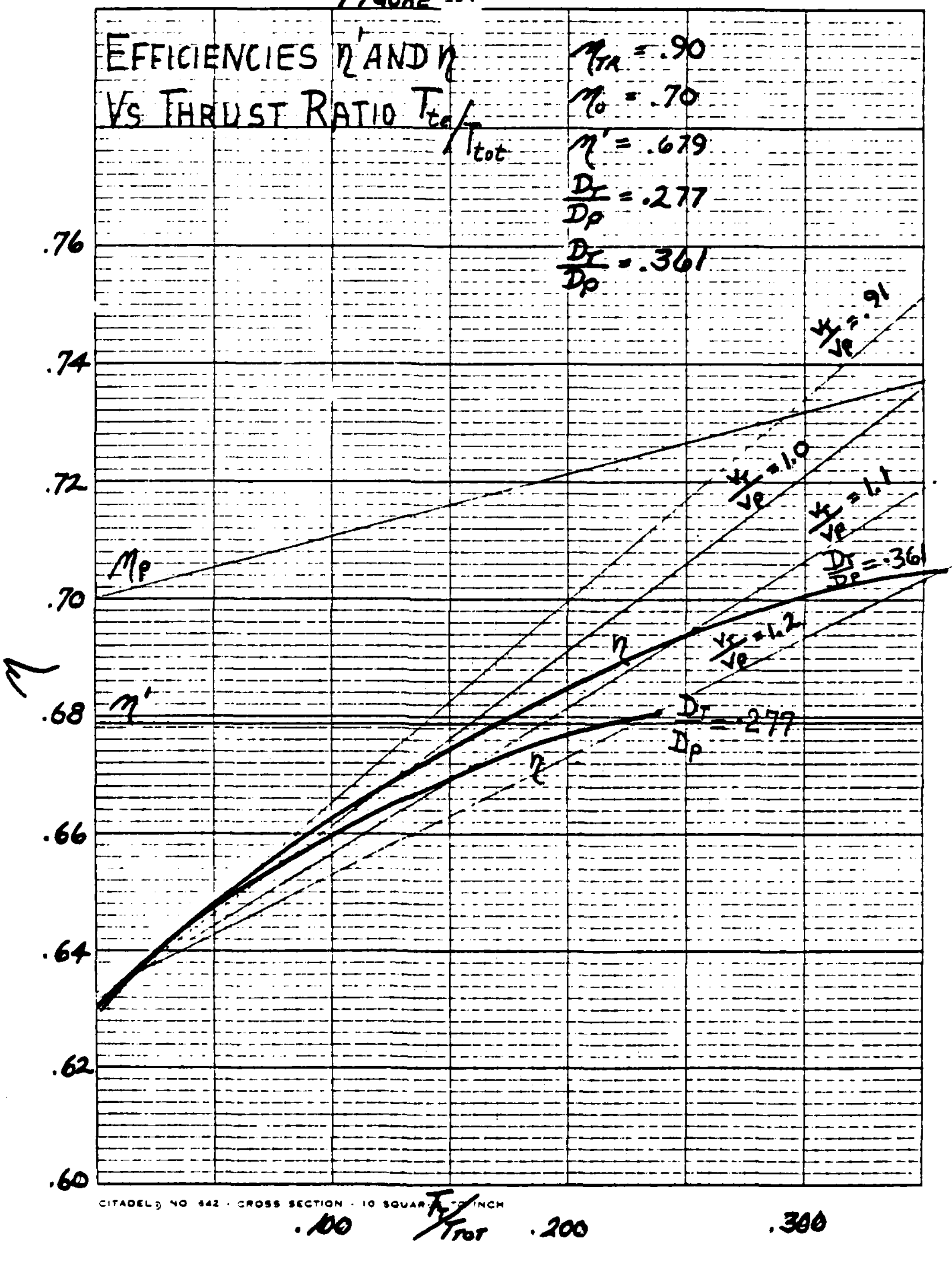

EFFICIENCIES η' AND η
VS THRUST RATIO T_{te}/T_{tot}
η_{TR} = .85
η_c = .70
η' = .679
D_T/D_P = .277
D_T/D_P = .361
.75
.73
.71
.69
.67
.65
.63
.61
.59
η
η_P
η'
V_T/V_P = .91
V_T/V_P = 1.0
V_T/V_P = 1.1
V_T/V_P = 1.2
D_T/D_P = .361
D_T/D_P = .277
CITADEL NO 442 · CROSS SECTION · 10 SQUARES TO INCH
.100
T_T/T_{TOT}
.200
.300

FIGURE 15.


EFFICIENCIES η' AND η
VS THRUST RATIO T_{te}/T_{tot}
$\eta_{TR} = .90$
$\eta_o = .70$
$\eta' = .679$
$\frac{D_T}{D_P} = .277$
$\frac{D_T}{D_P} = .361$
.76
.74
.72
.70
.68
.66
.64
.62
.60
η_P
η'
η
$V_c/V_e = .91$
$V_c/V_e = 1.0$
$V_c/V_e = 1.1$
$V_c/V_e = 1.2$
$\frac{D_T}{D_P} = .361$
$\frac{D_T}{D_P} = .277$
CITADEL NO 442 · CROSS SECTION · 10 SQUARES TO INCH
.100
.200
.300
T_T/T_{TOT}

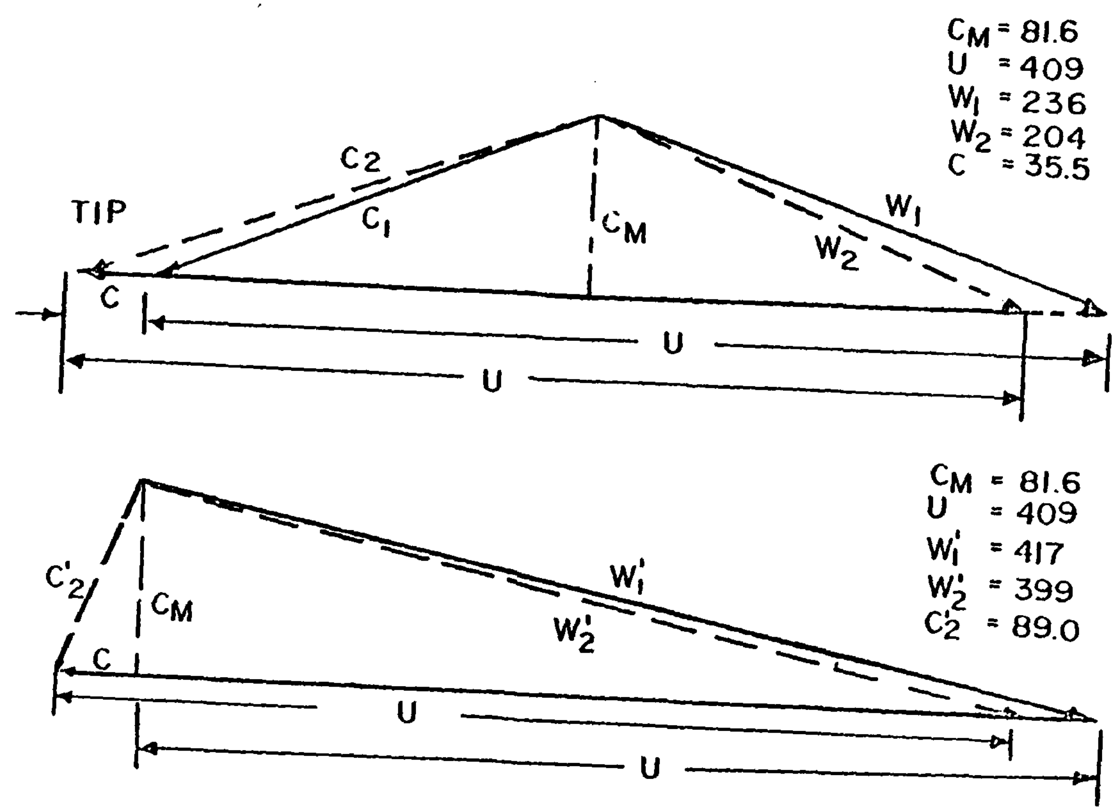
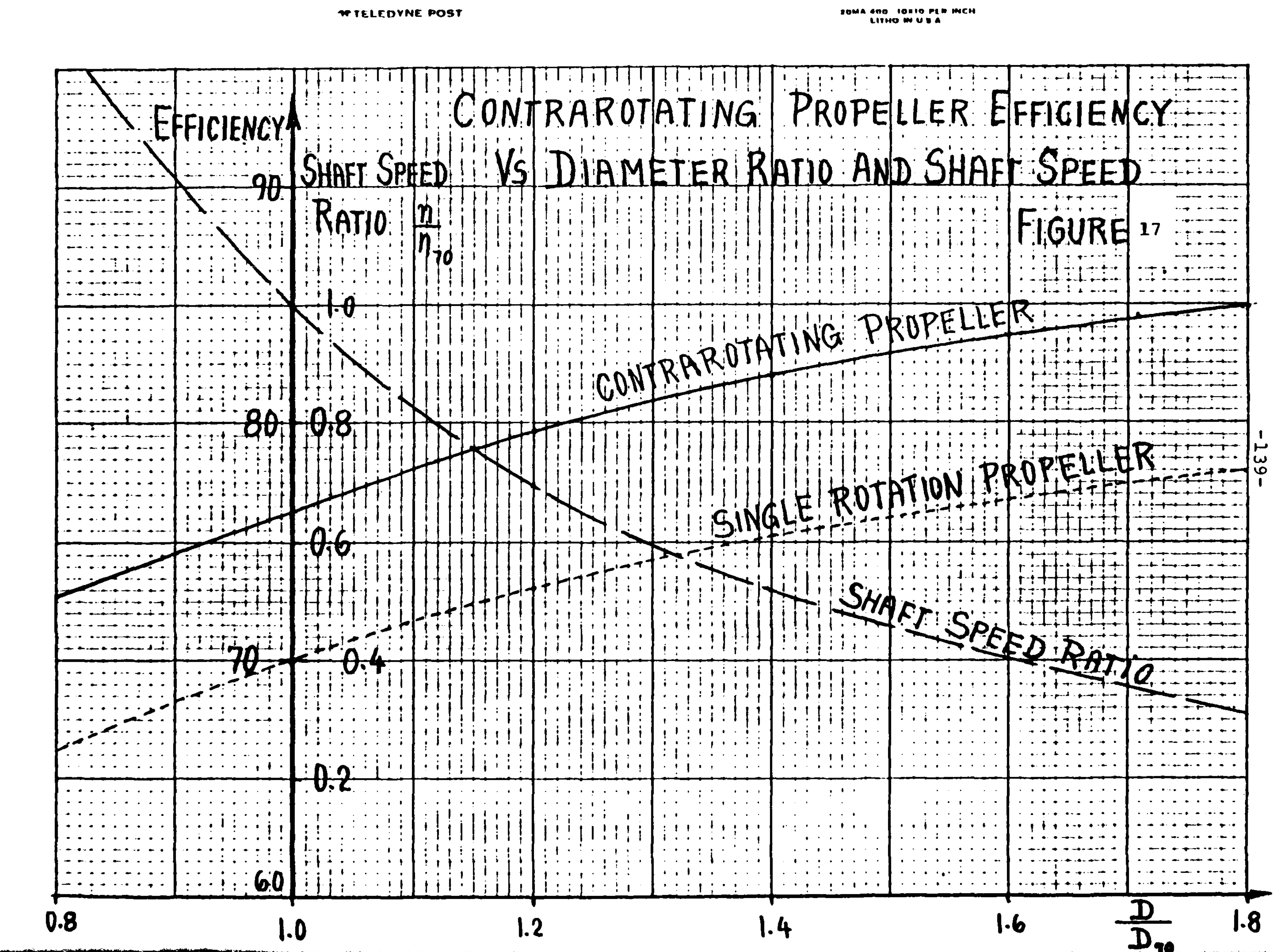


FIGURE 16. Flow Vector Diagram at Impeller Tip for 50% Reaction Pump (Top) and Standard Pump (Bottom)


TELEDYNE POST
20MA 400 10X10 PER INCH LITHO IN U.S.A.
CONTRAROTATING PROPELLER EFFICIENCY
VS DIAMETER RATIO AND SHAFT SPEED
FIGURE 17
EFFICIENCY
SHAFT SPEED
RATIO $\frac{n}{n_{70}}$
90
80
70
60
1.0
0.8
0.6
0.4
0.2
CONTRAROTATING PROPELLER
SINGLE ROTATION PROPELLER
SHAFT SPEED RATIO
0.8
1.0
1.2
1.4
1.6
1.8
$\frac{D}{D_{70}}$

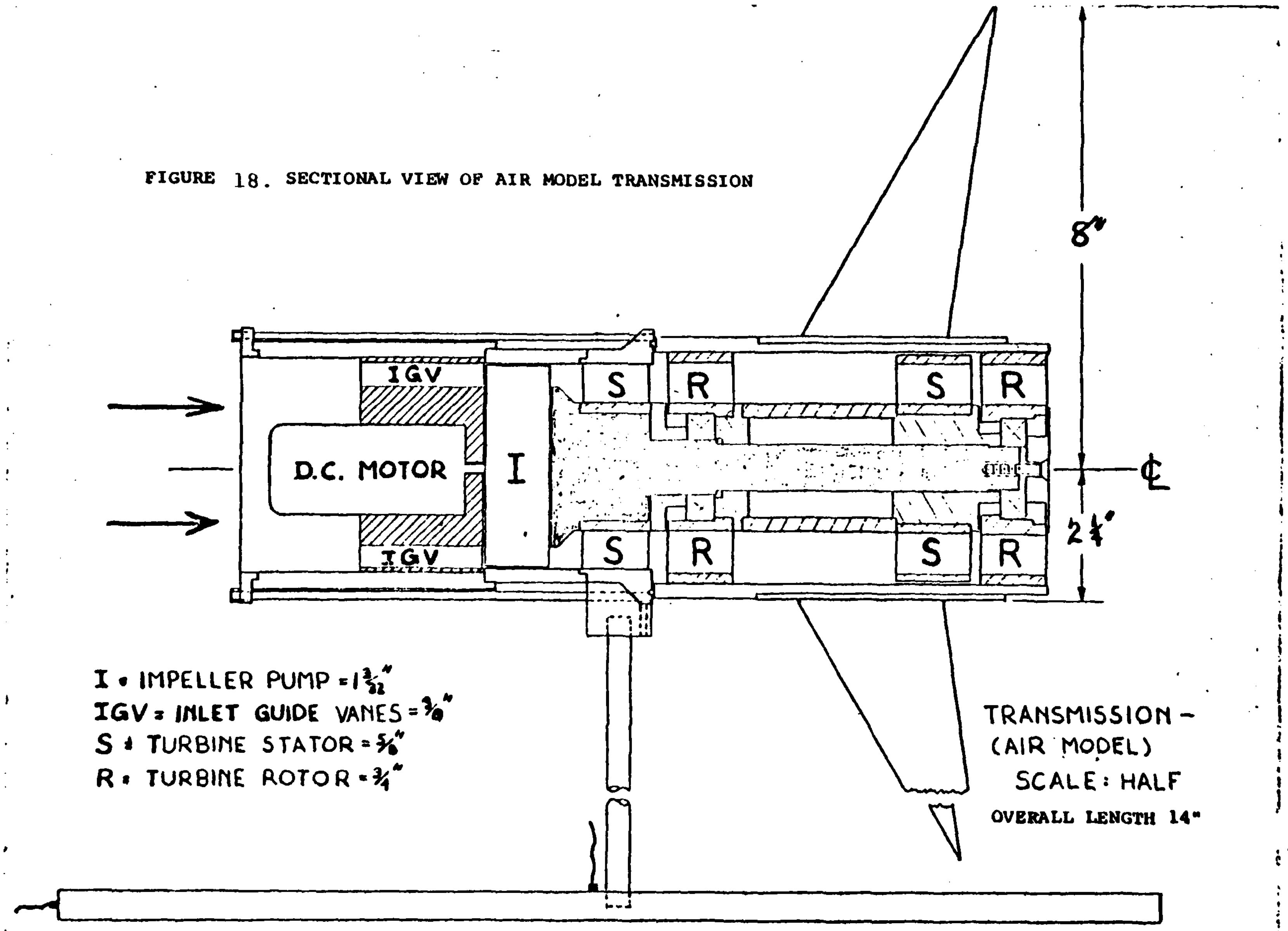


FIGURE 18. SECTIONAL VIEW OF AIR MODEL TRANSMISSION

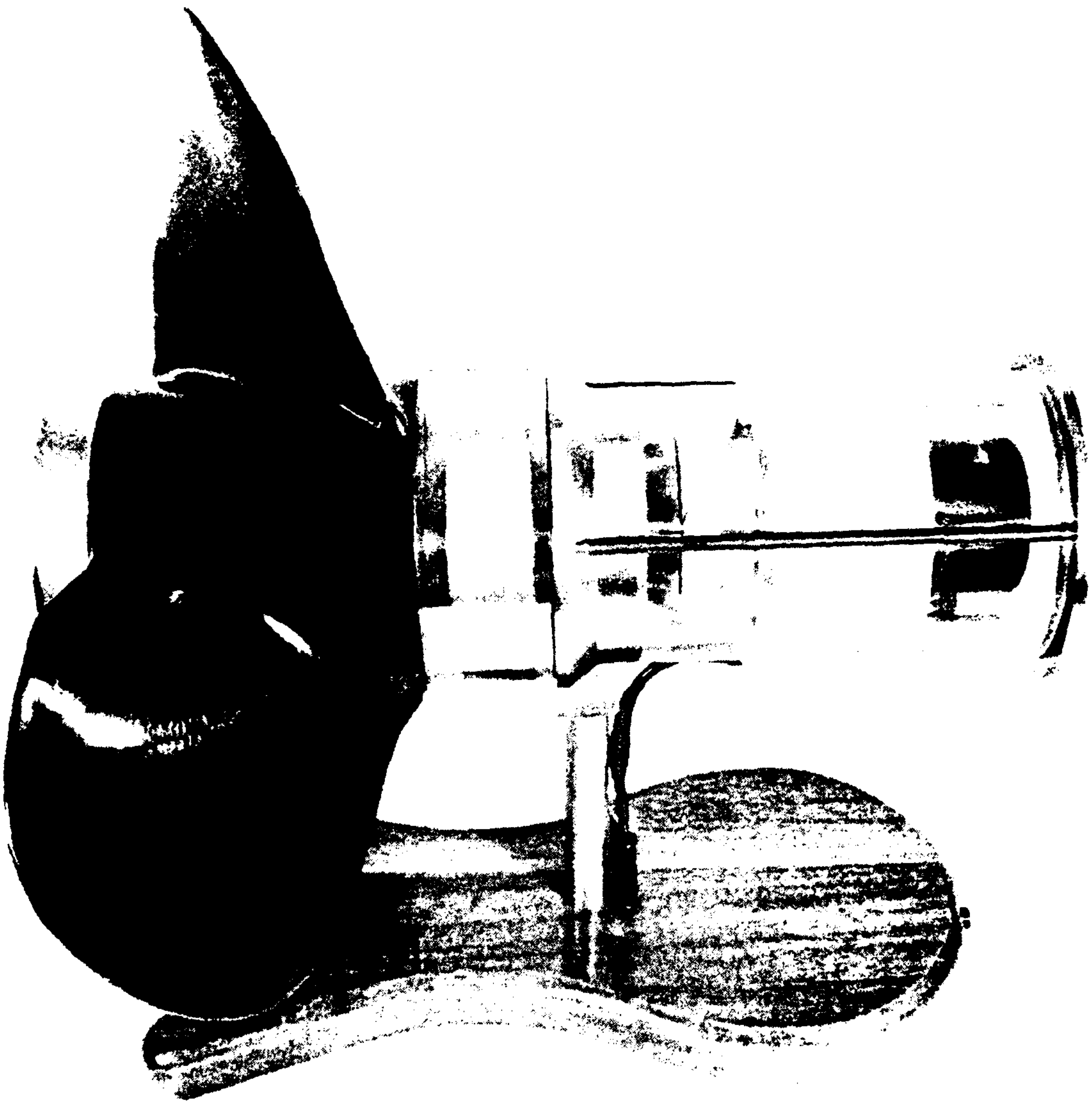


FIGURE 19. Assembled Air Model Transmission

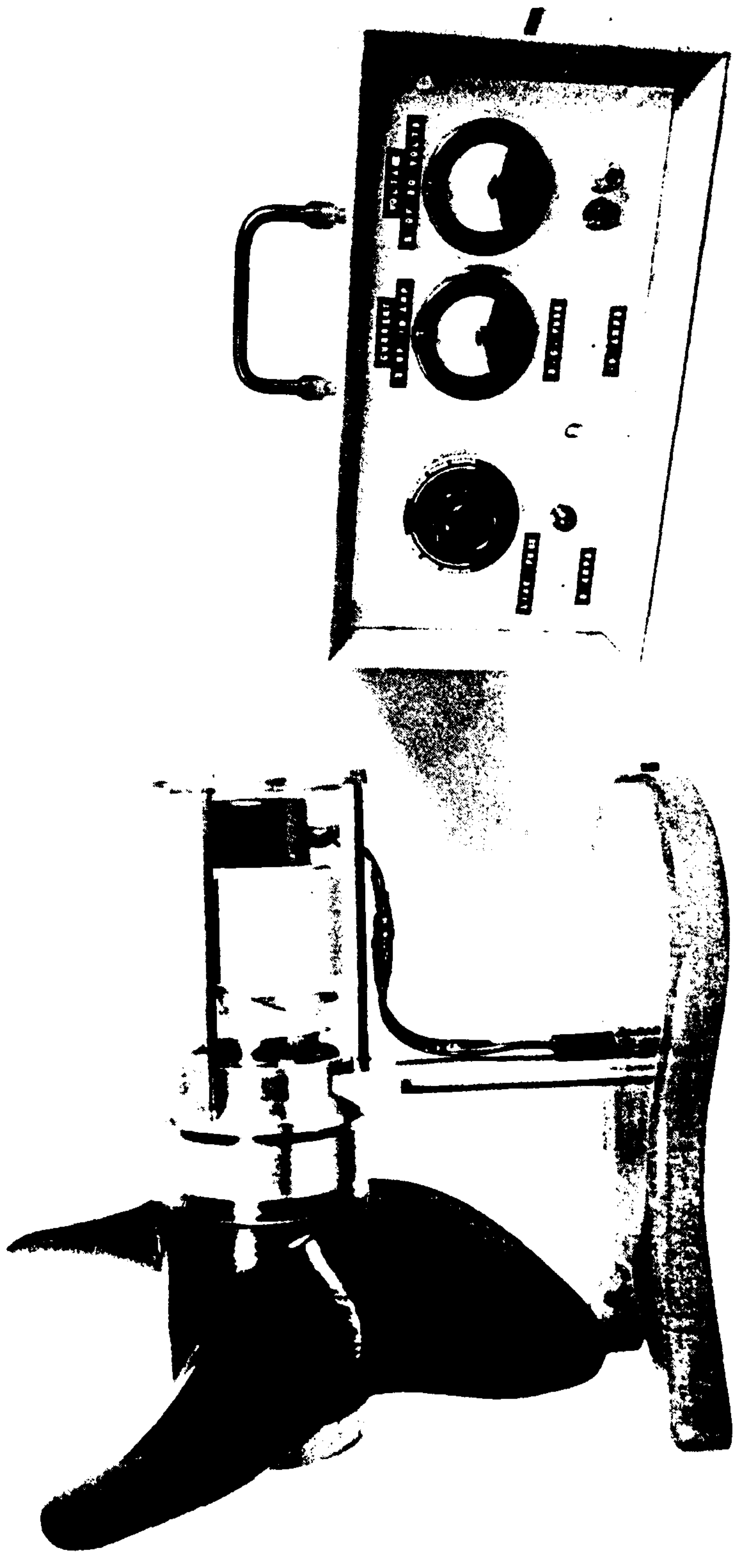


FIGURE 20. Air Model Transmission and Power Supply

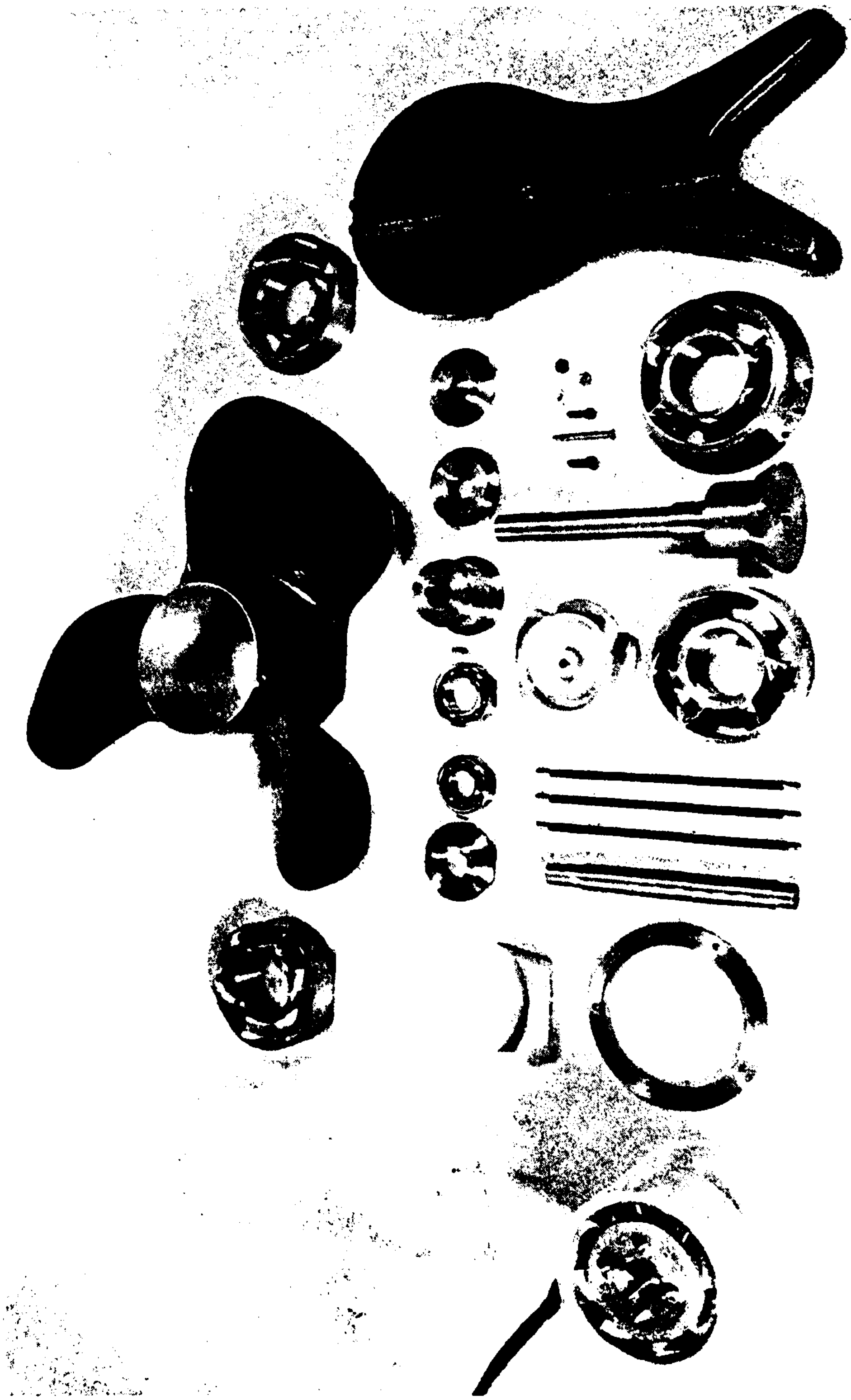


FIGURE 21. Air Model Transmission Component Parts

FIGURE 22. Air Model Transmission, Impeller Speed Versus Turbine Speed

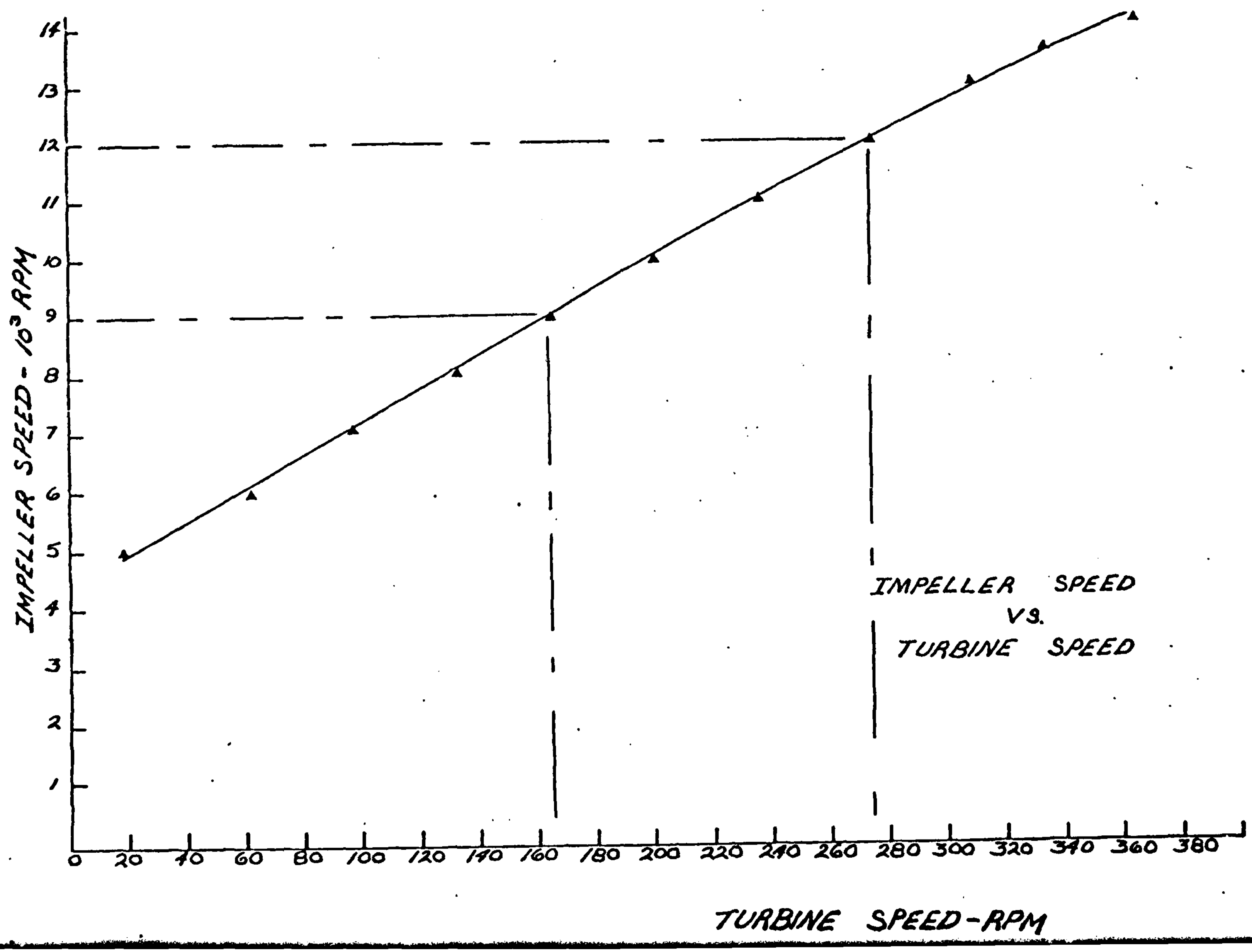
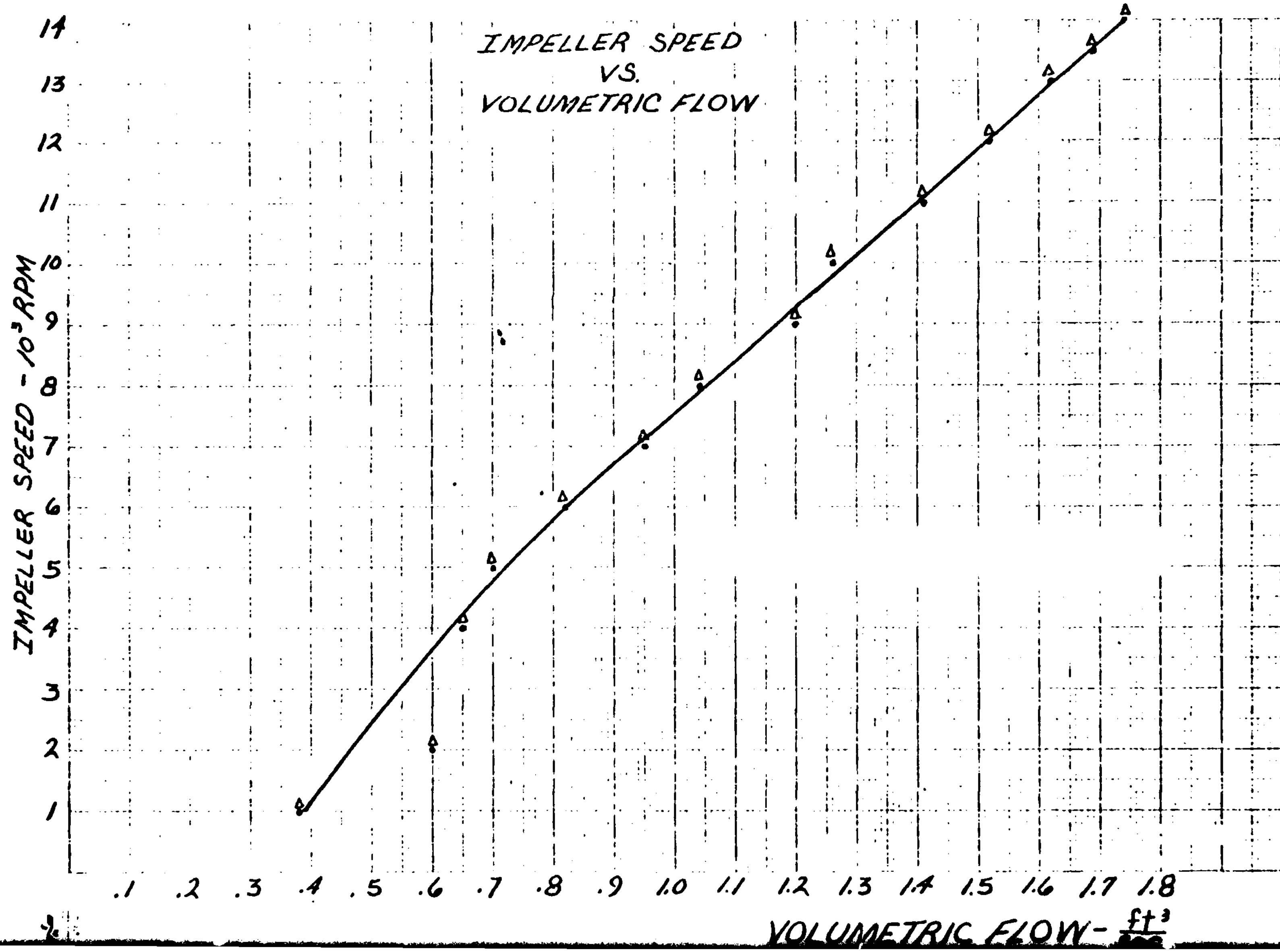


FIGURE 23. Air Model Transmission, Impeller Speed Versus Turbine Speed



I - IMPELLER, TIP DIA. 2.70"
EGV - EXIT GUIDE VANES, TIP DIA. 2.70"
S_r - R.H. STATOR RING, TIP DIA. 3.75"
R_r - R.H. ROTOR RING, TIP DIA. 3.75"
S_l - L.H. STATOR RING, TIP DIA. 3.73"
R_l - L.H. ROTOR RING, TIP DIA. 3.75"
P_r - R.H. PROPELLER, TIP DIA. 15.0"
P_l - L.H. PROPELLER, TIP DIA. 15.0"

OVERALL LENGTH 14.0" ---- SCALE 1:1.54

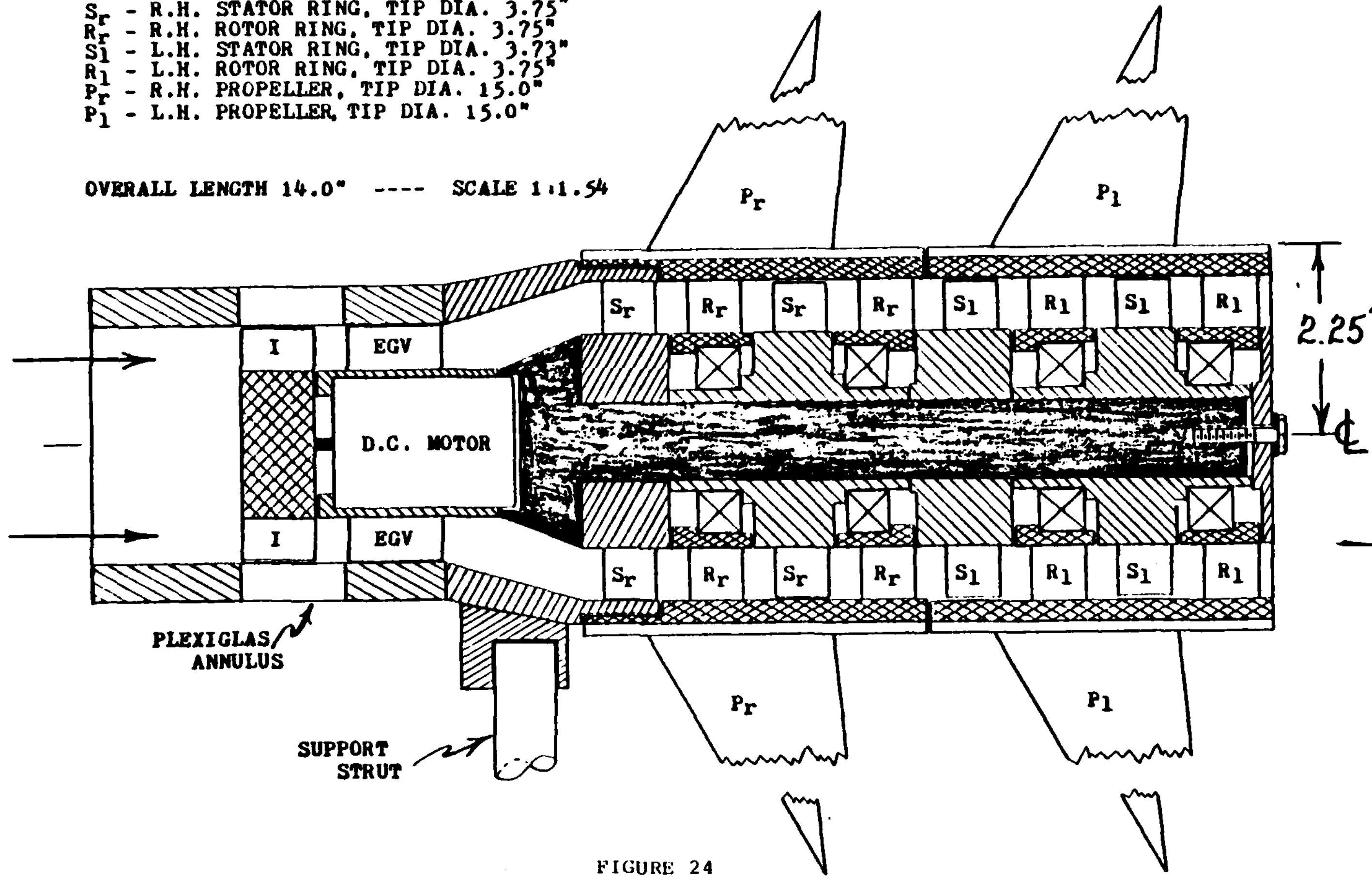


FIGURE 24
SECTIONAL VIEW OF CONTRA ROTATING AIR MODEL

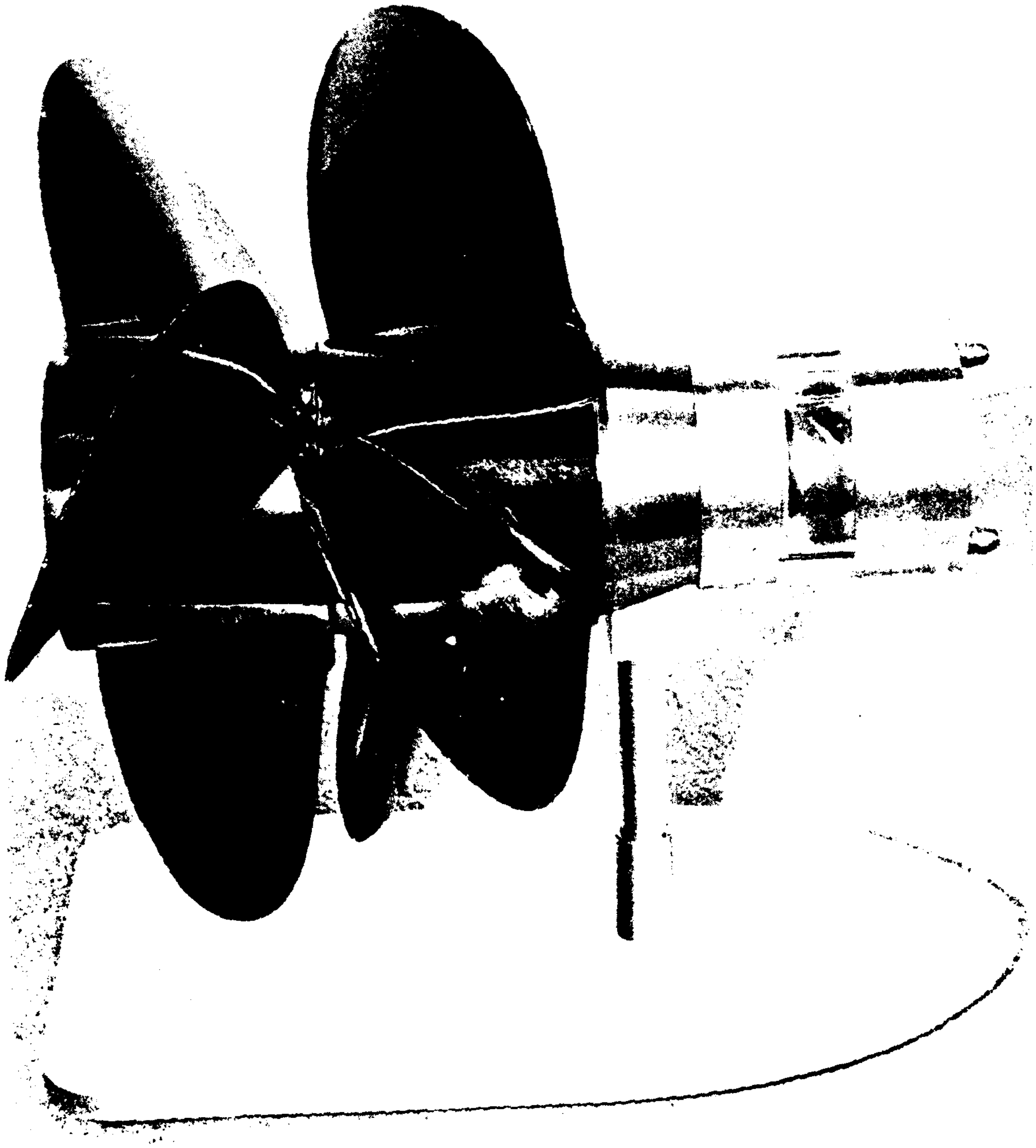


FIGURE 25. Contra Rotating Air Model

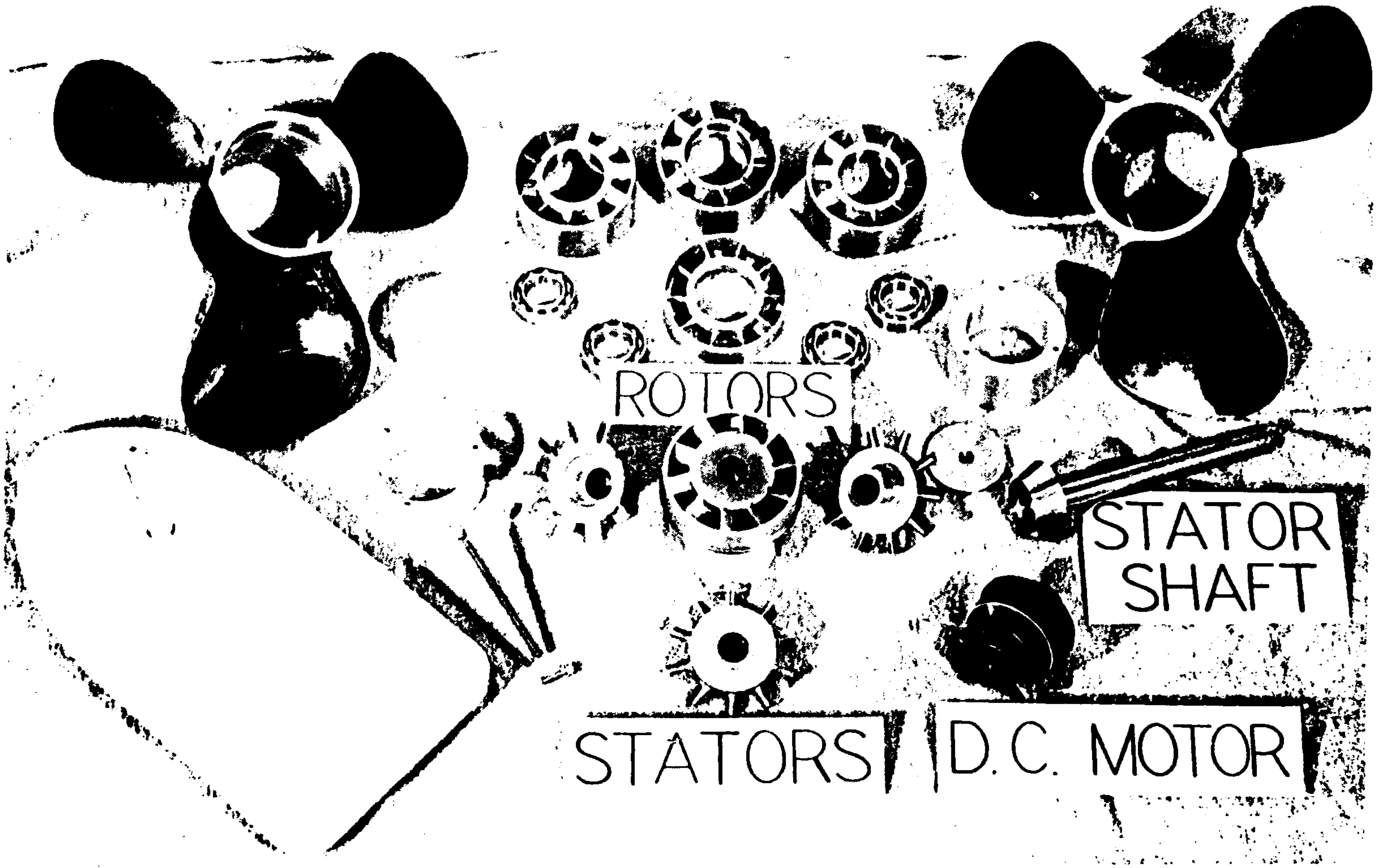


FIGURE 26. Component Parts of Contra Rotating Air Model

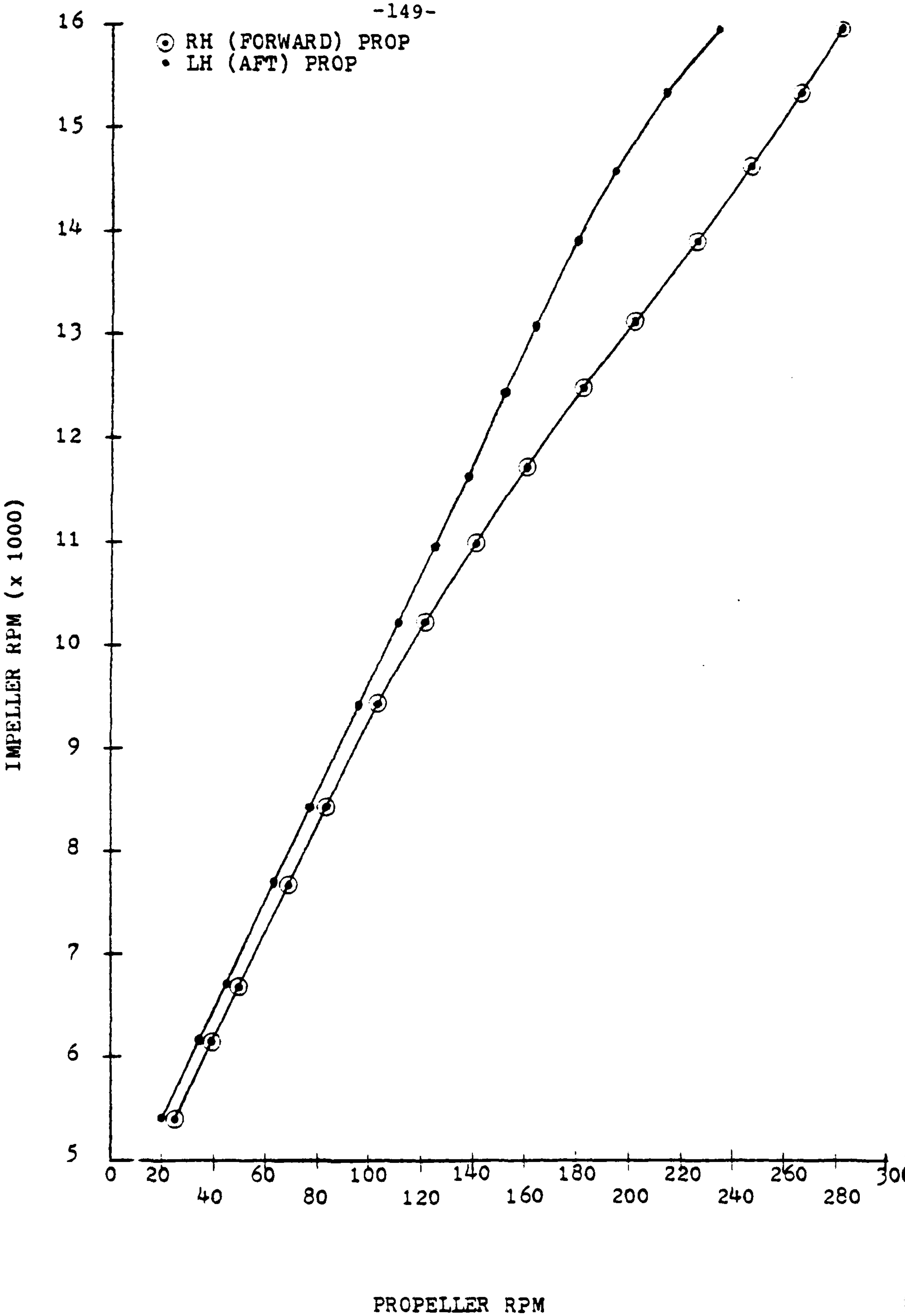


FIGURE 27a. Contra Rotating Air Model Propeller Versus Impeller Speed

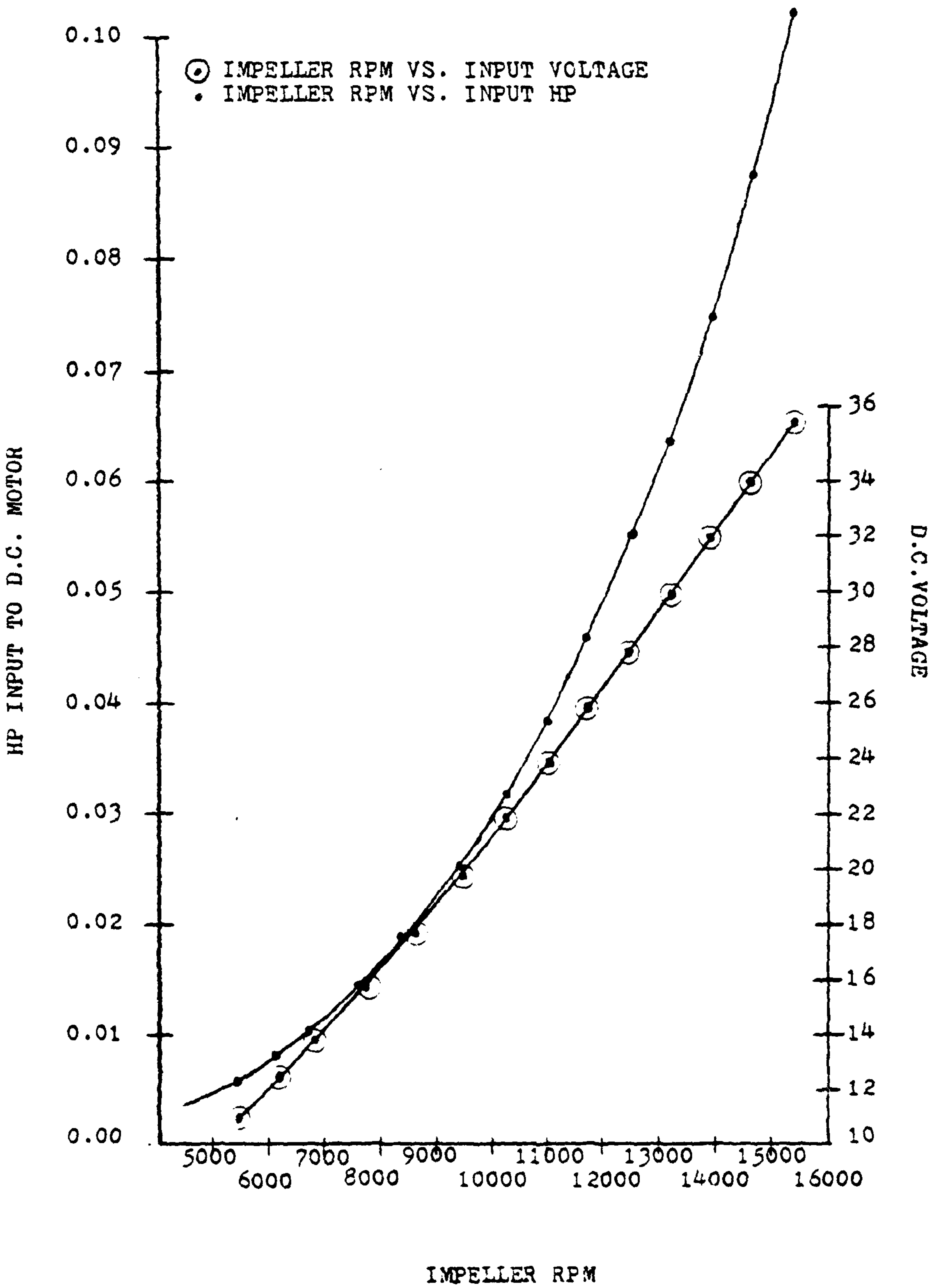
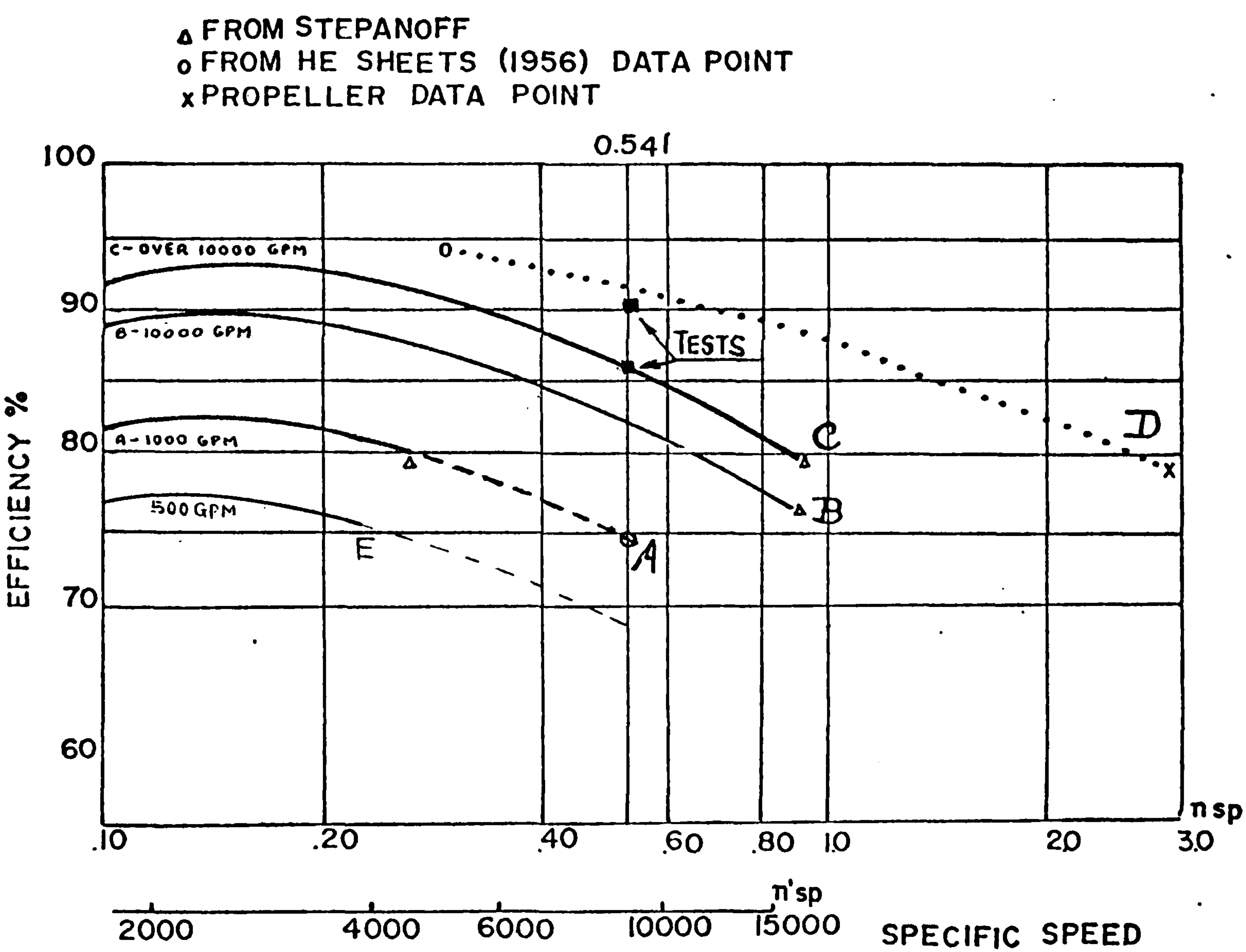


FIGURE 27b. *Contra Rotating Air Model Fan Horsepower Versus Impeller Speed*

FIGURE 28.

PUMP EFFICIENCY vs SPECIFIC SPEED



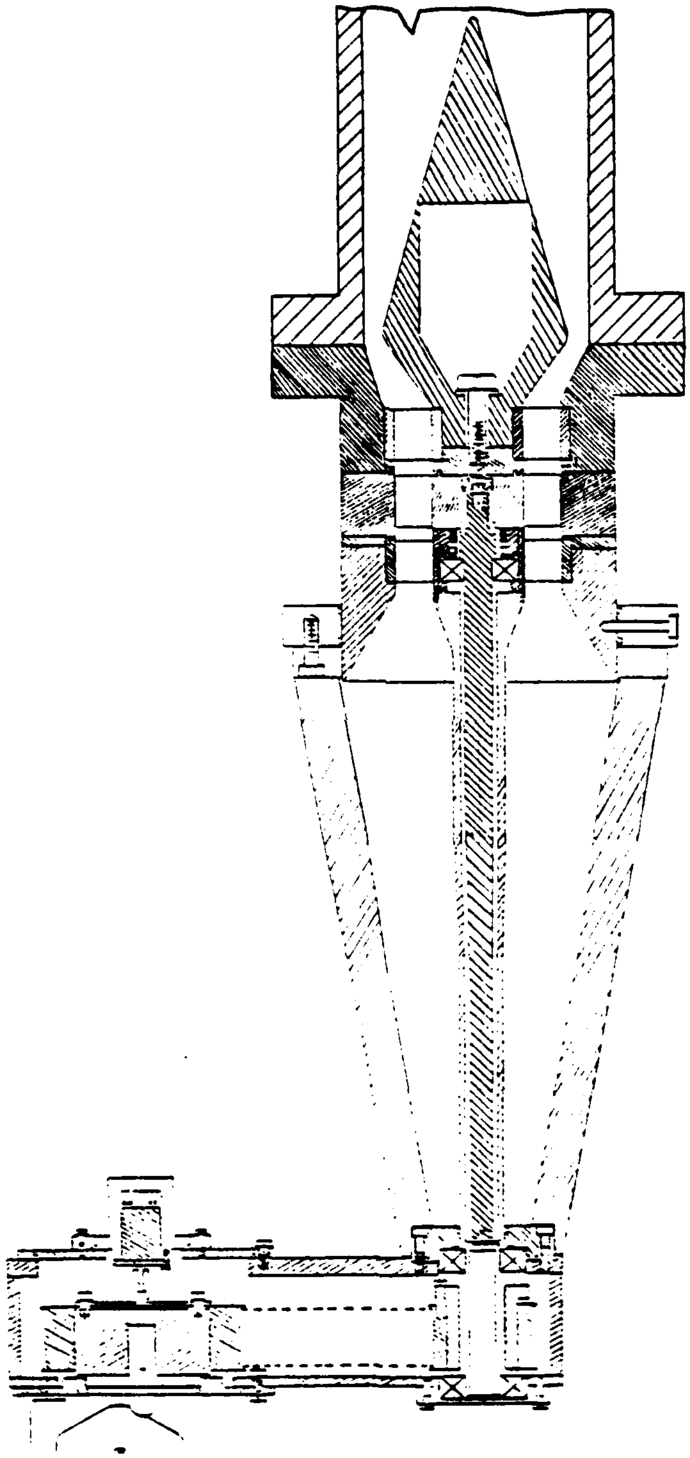


FIGURE 29.
Pump Design

FIGURE 30. Pump Impeller, Inlet Vane Ring, Exit Vane Ring

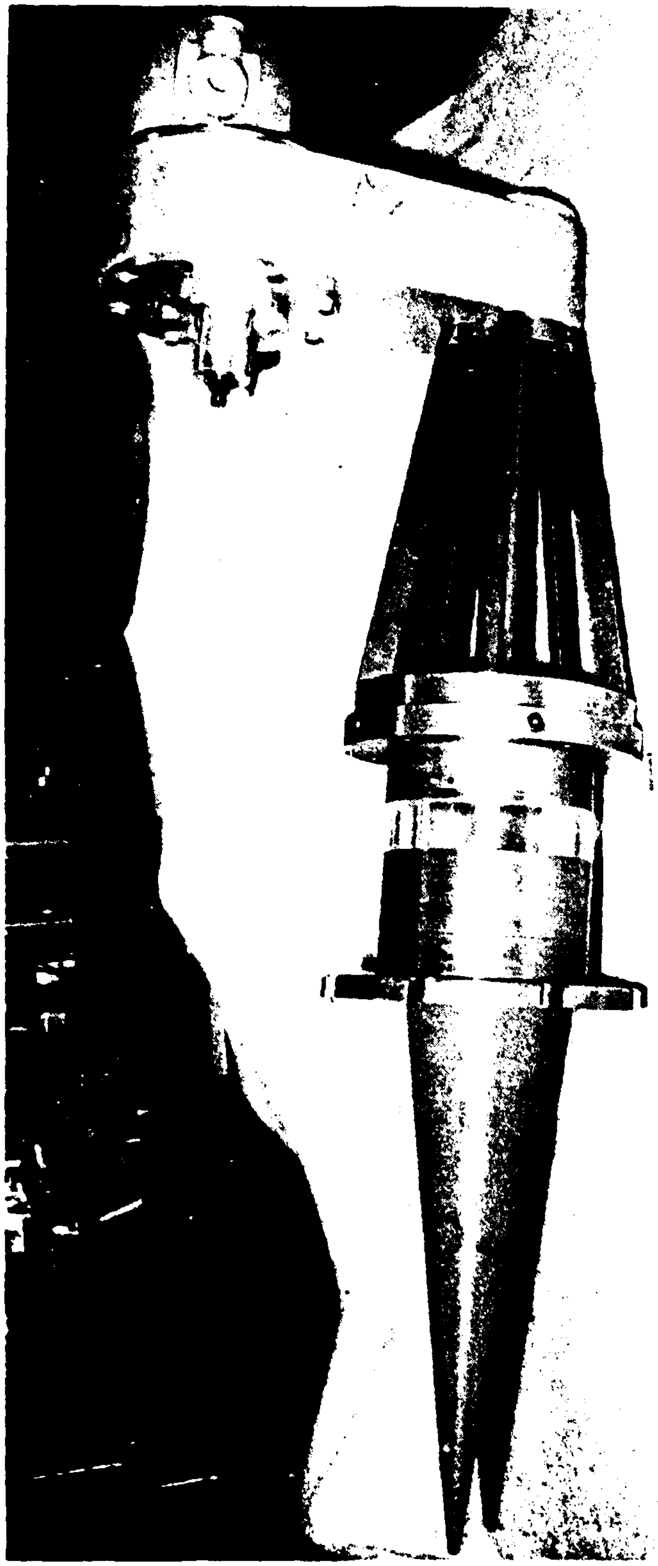
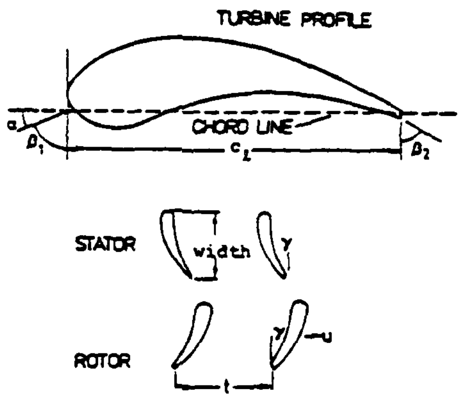


FIGURE 31. Transfer Case and Assembled Pump



29 blade stators 28 blade rotors

width (in)	t (in)	c_ℓ (in)	s	β_1 (deg)	β_2 (deg)	γ (deg)	
.583	.547	.613	1.12	72.00	59.70	18.00	1st stage stator
.560	.547	.613	1.12	83.80	48.22	24.00	the rest of the stators
.560	.567	.613	1.08	83.80	48.22	24.00	all the rotors

D_{tip} = 5.7 in. , D_{mean} = 5.06 in. , D_{hub} = 4.4 in.

FIGURE 32.

Turbine Blade Specifications

FIGURE 33.

FLOW VECTOR DIAGRAM TURBINE 23-T 7-3

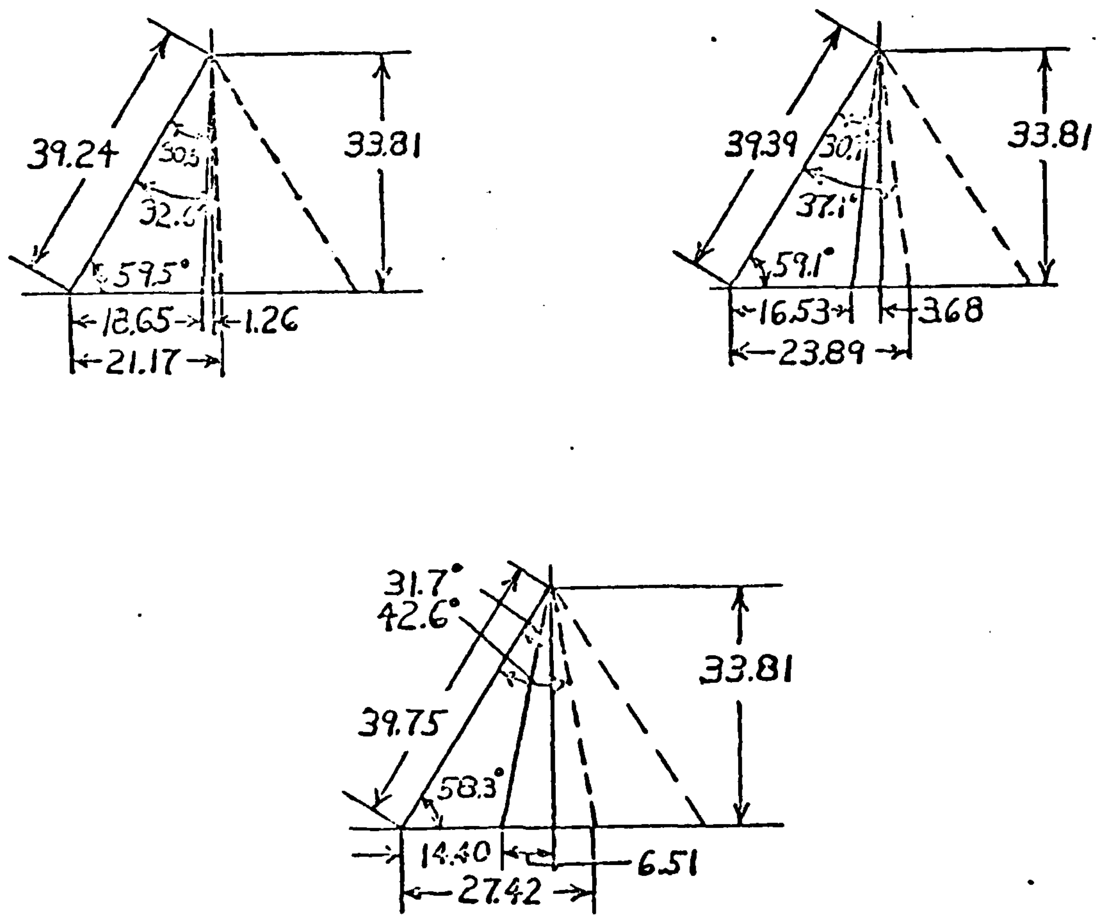
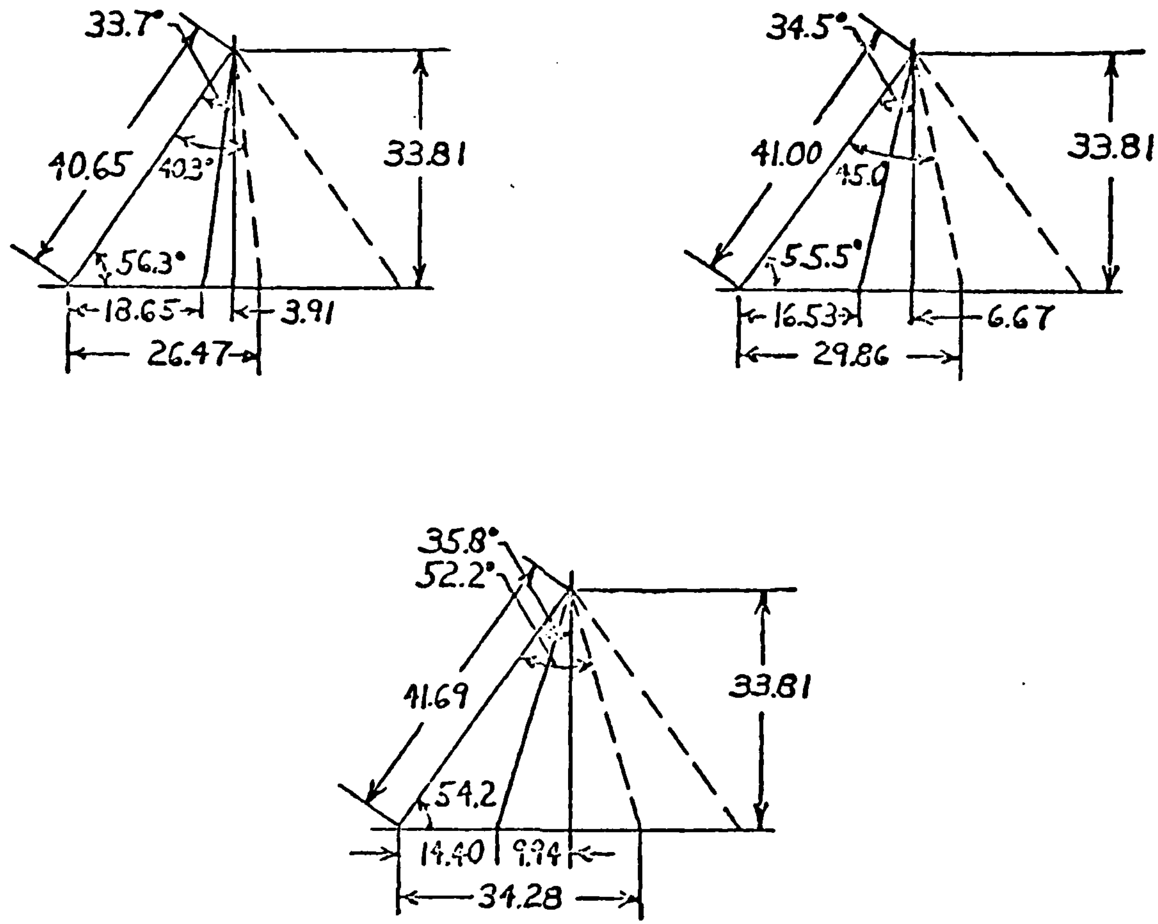


FIGURE 34.

FLOW VECTOR DIAGRAM TURBINE 24-T 7-4



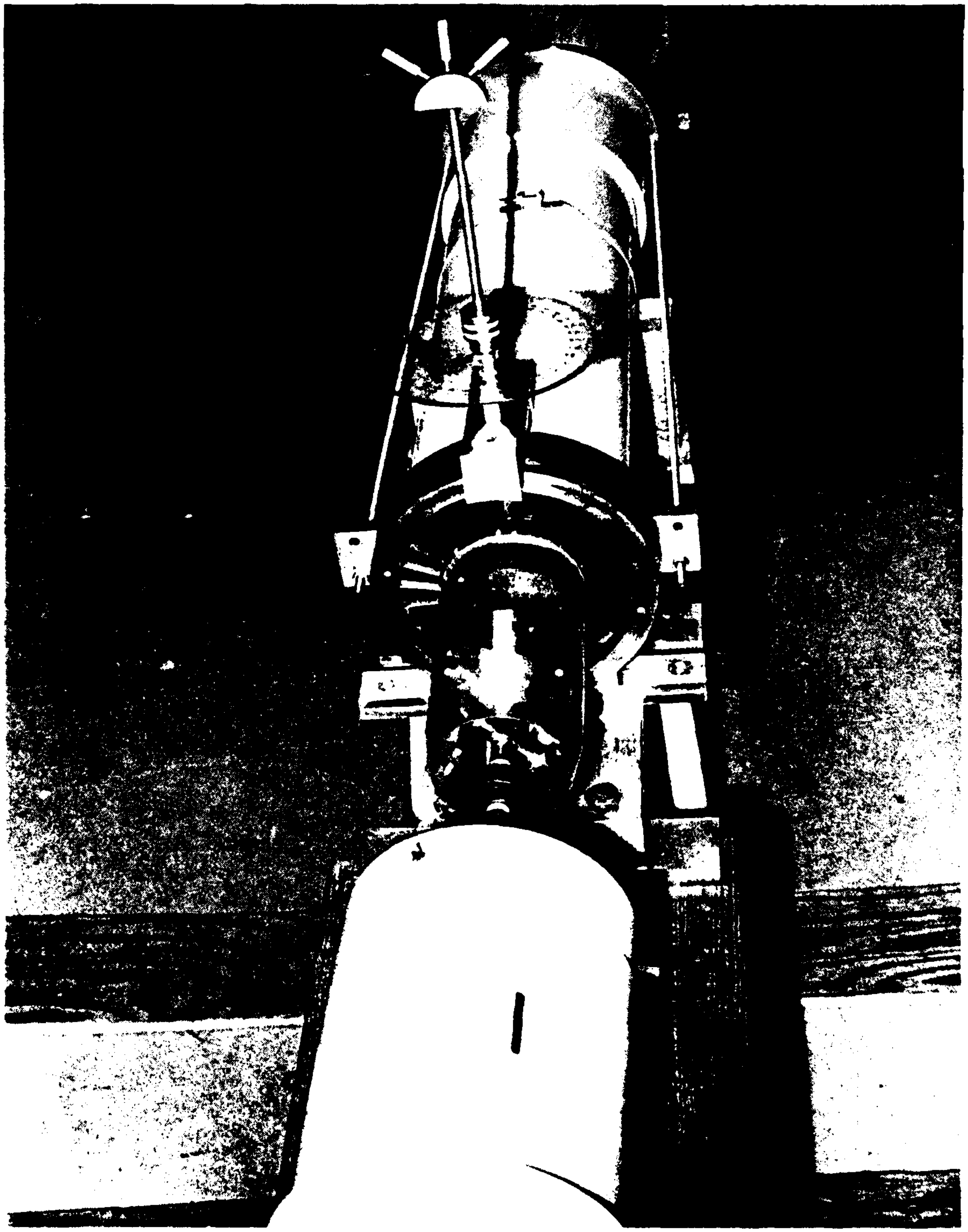
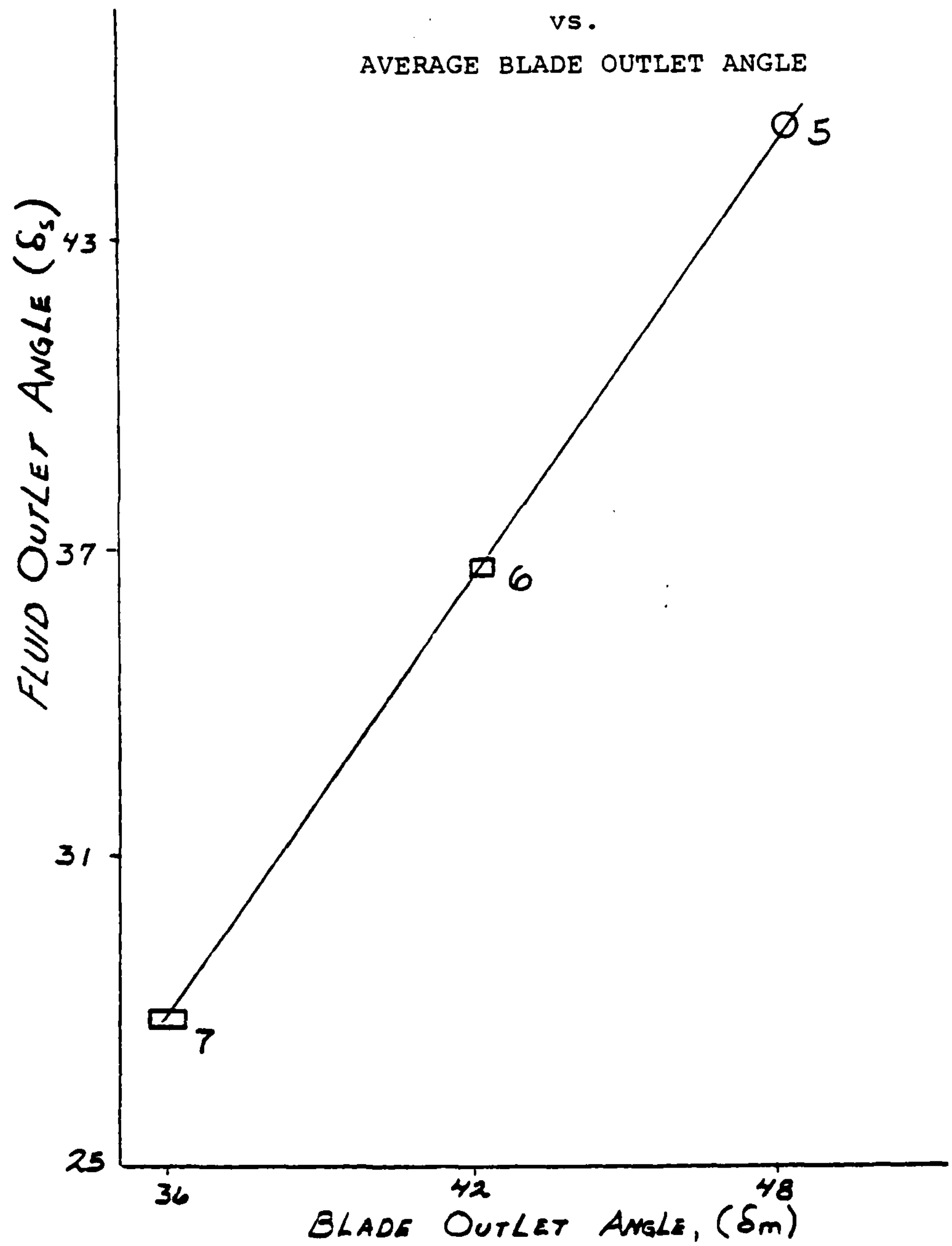


FIGURE 35. Cascade Test Arrangement

FIGURE 36.

CASCADE TESTS

AVERAGE FLUID OUTLET ANGLE
vs.
AVERAGE BLADE OUTLET ANGLE



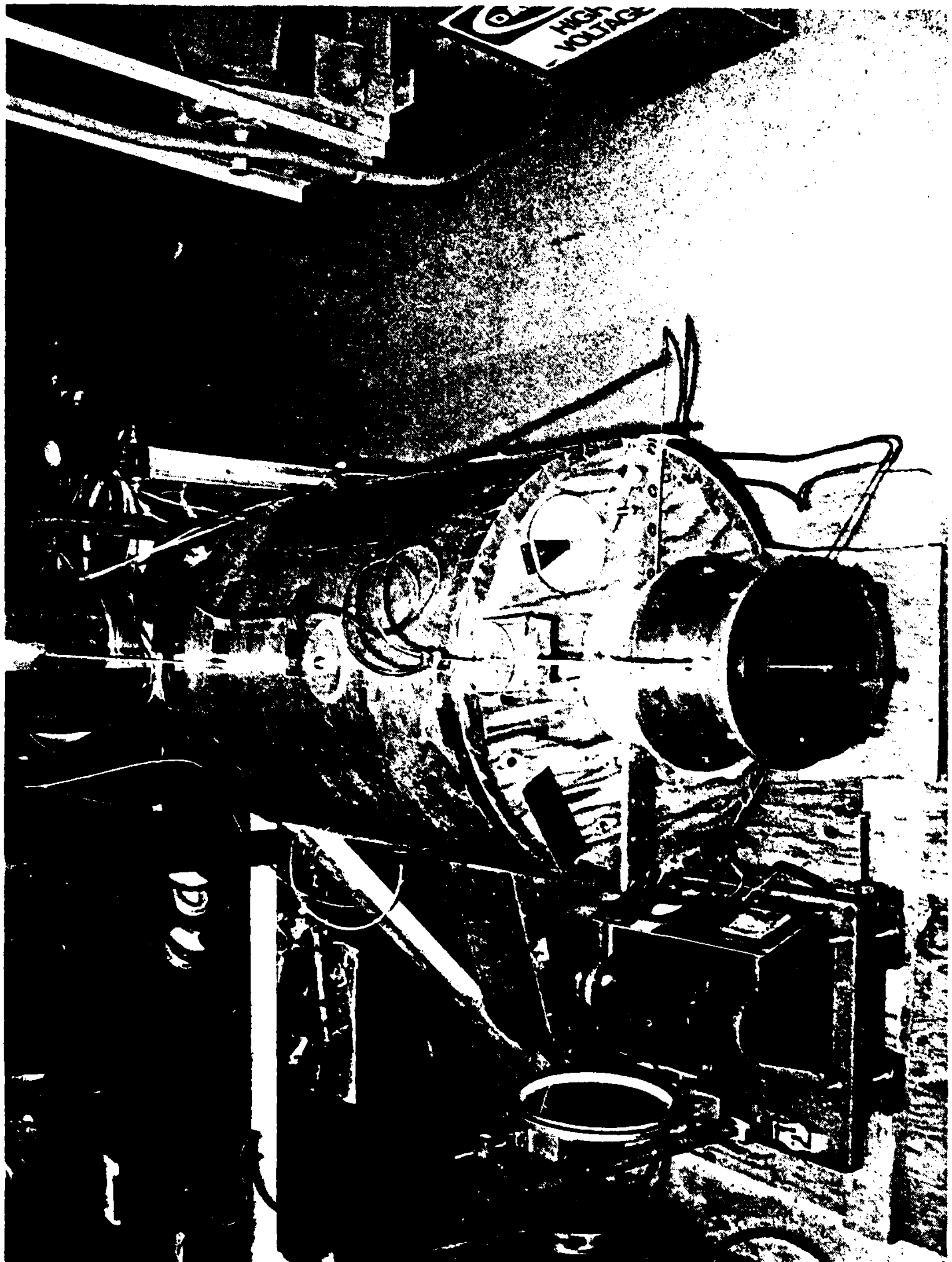


FIGURE 37. Air Turbine Test Arrangement

FIGURE 38.

Air Turbine Performance

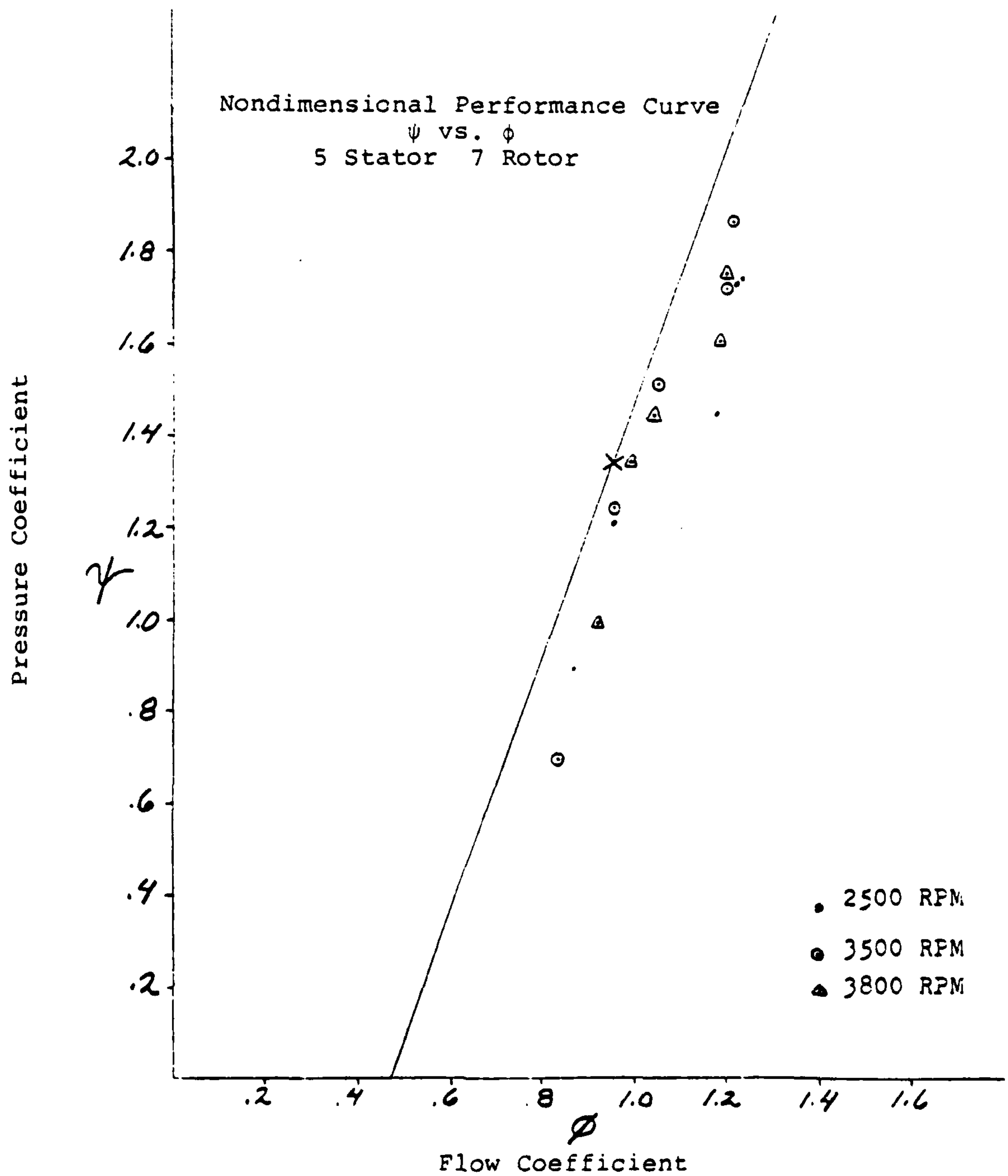
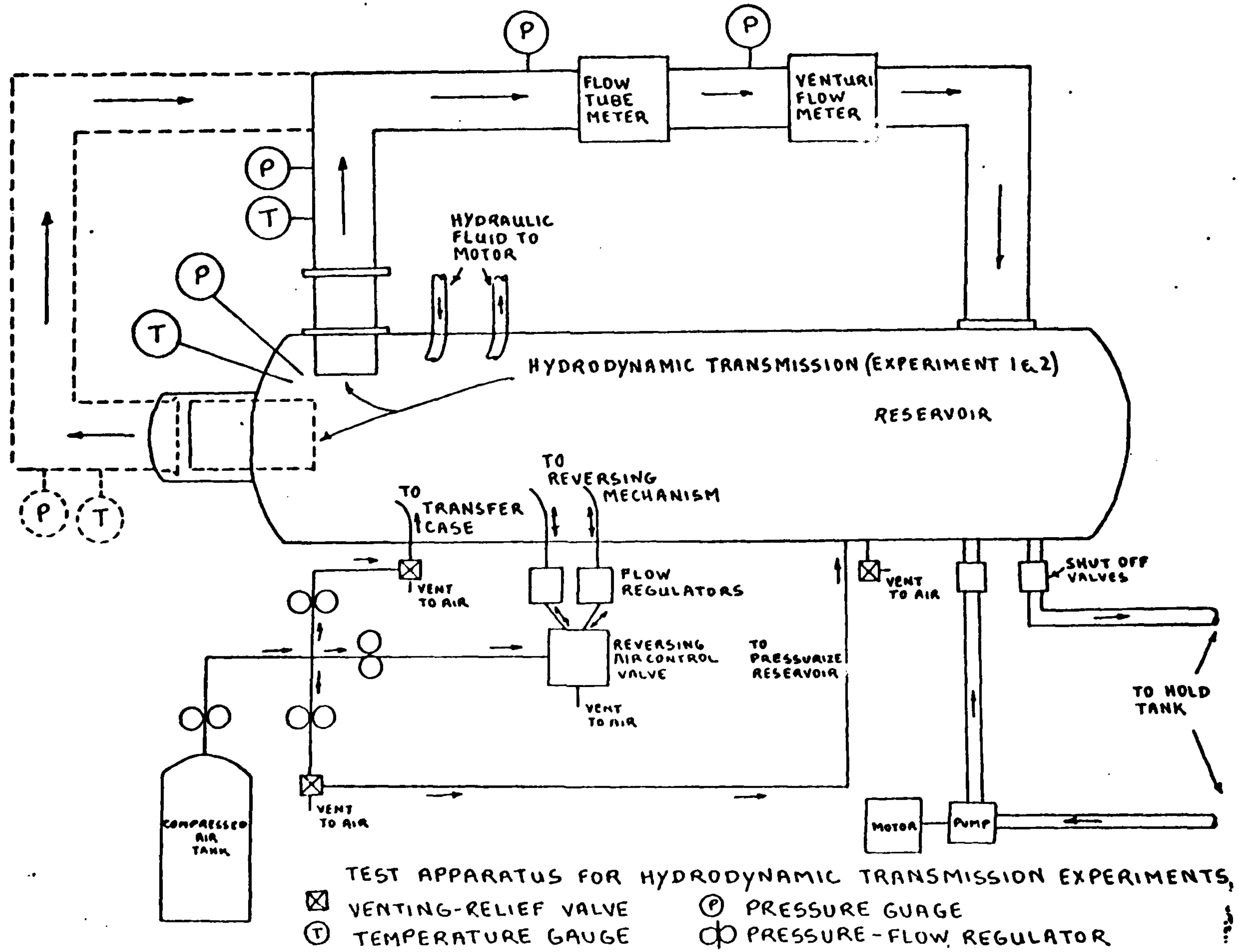


FIGURE 39. Water Test Loop for Hydraulic Transmission



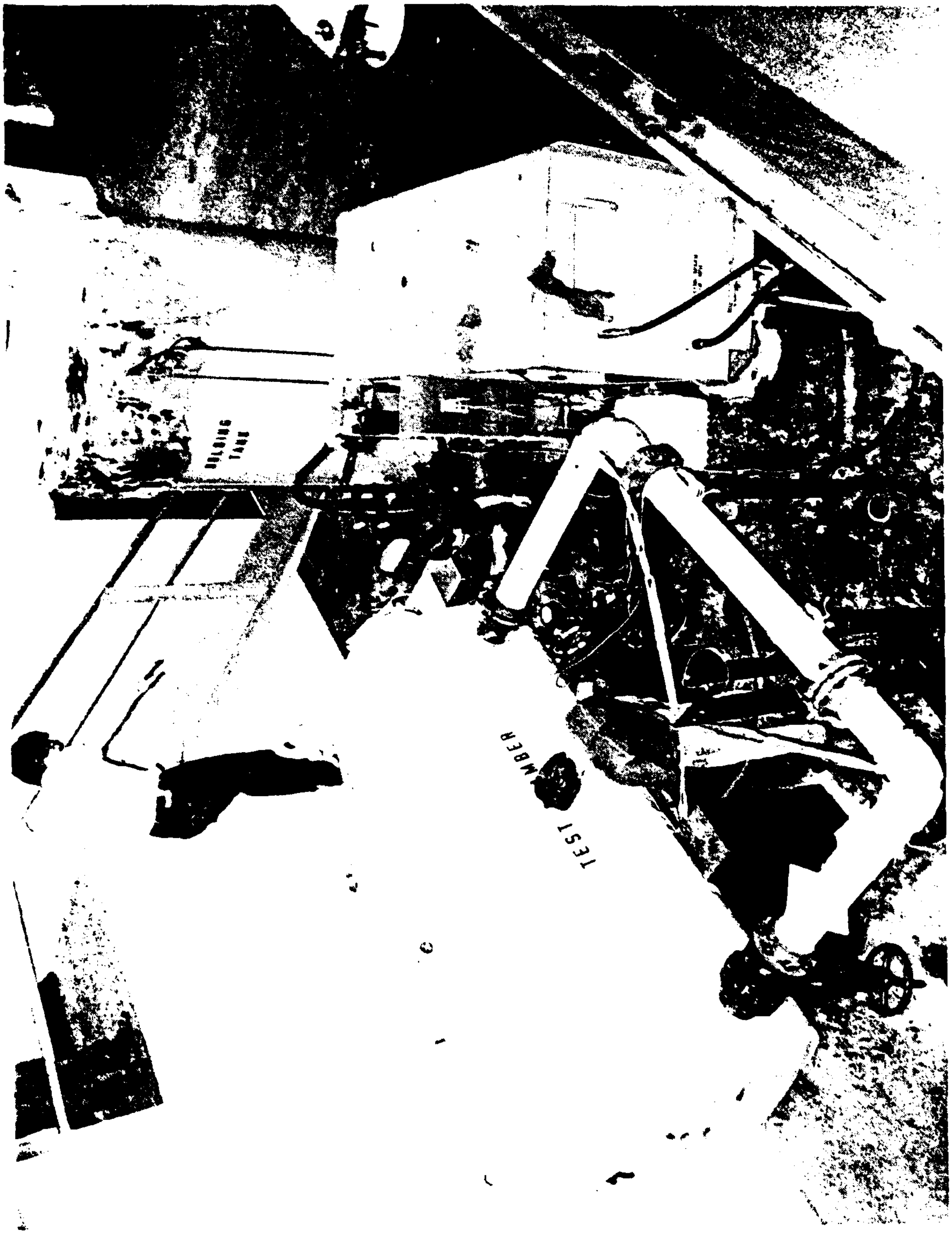


FIGURE 40. Test Loop and Tanks

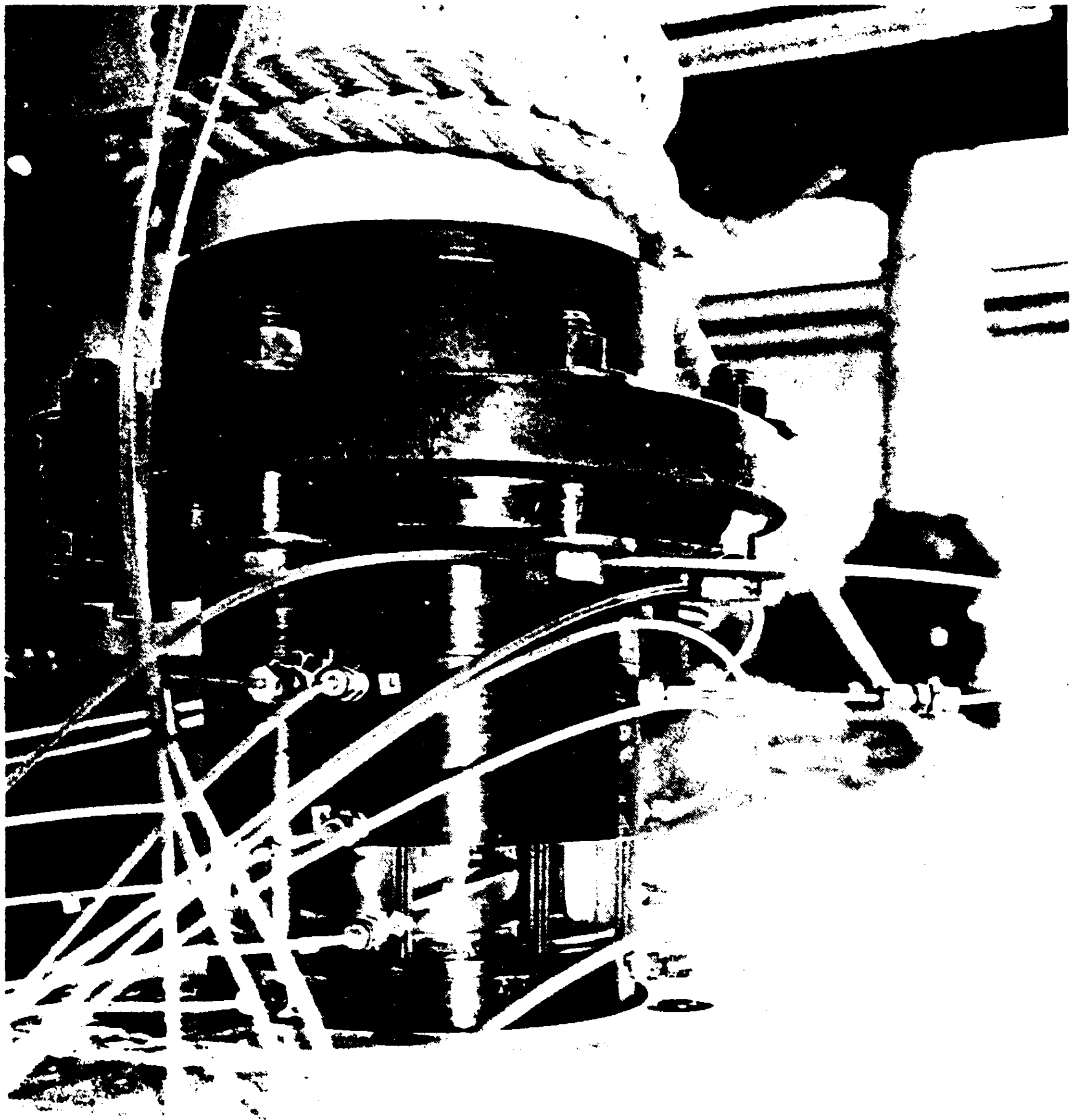


FIGURE 41. Pump and Test Instrumentation

FIGURE 42.

Pump #1--Performance

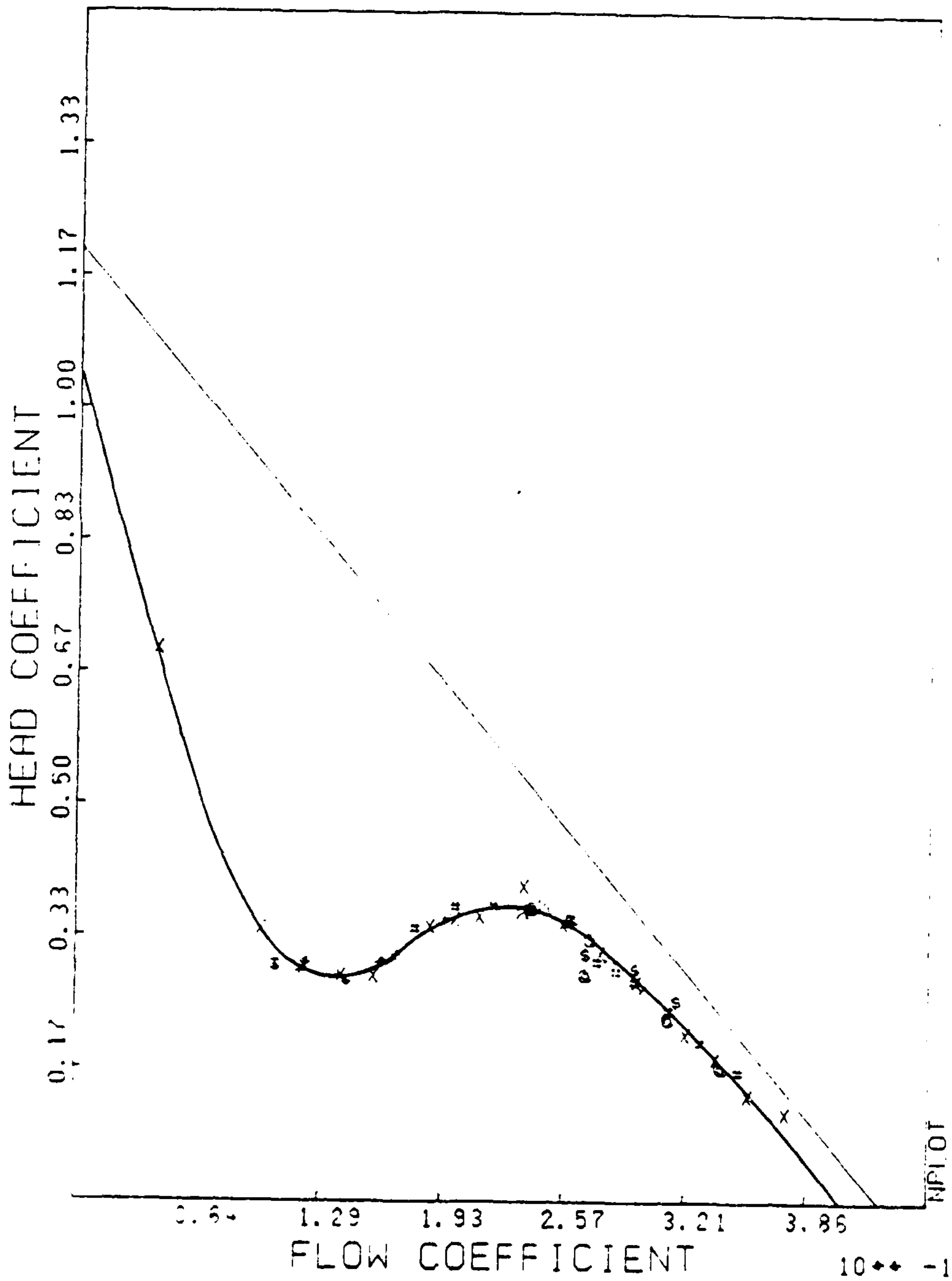


FIGURE 43.

Pump #2--Performance
Total Efficiency

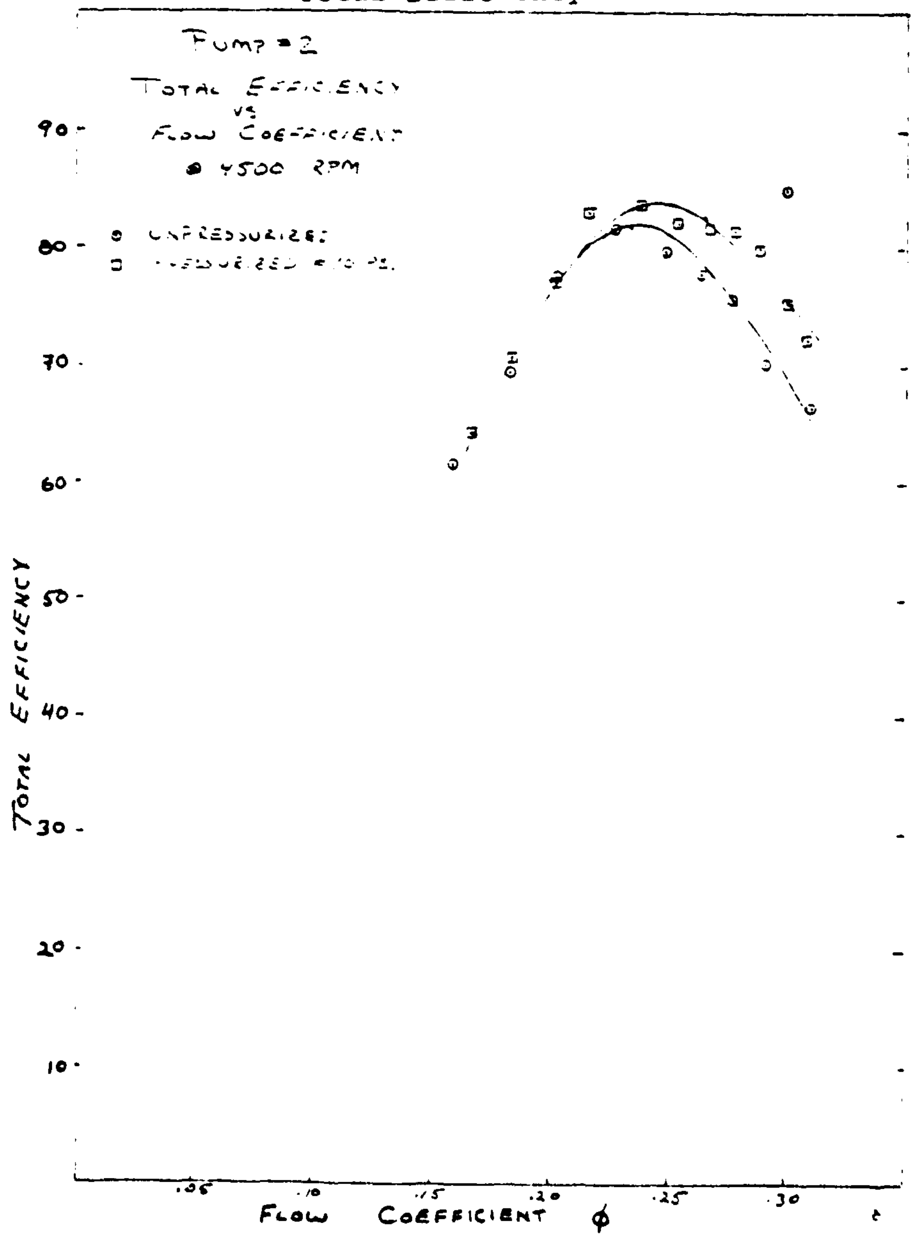


FIGURE 44.

Pump #2--Performance

Stage Efficiency

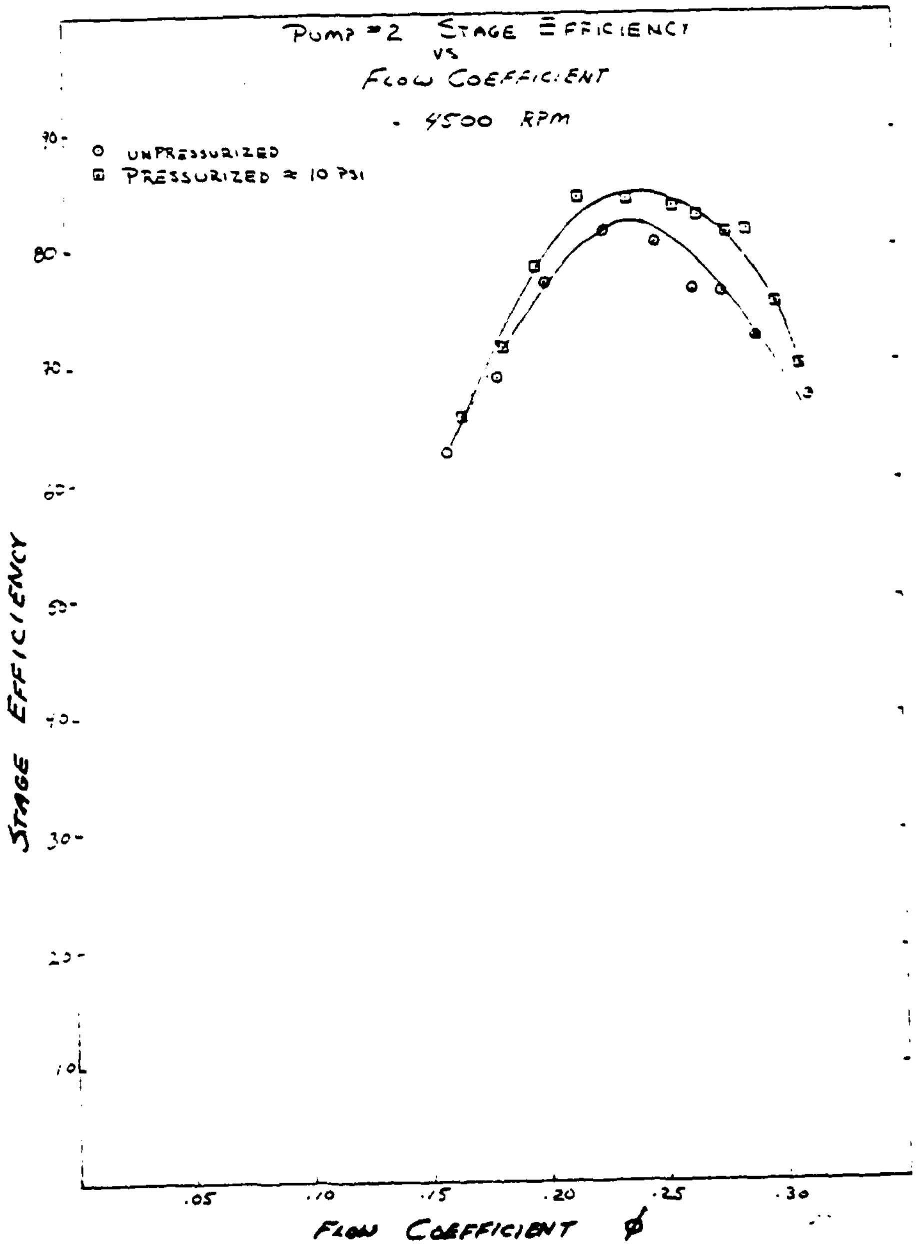
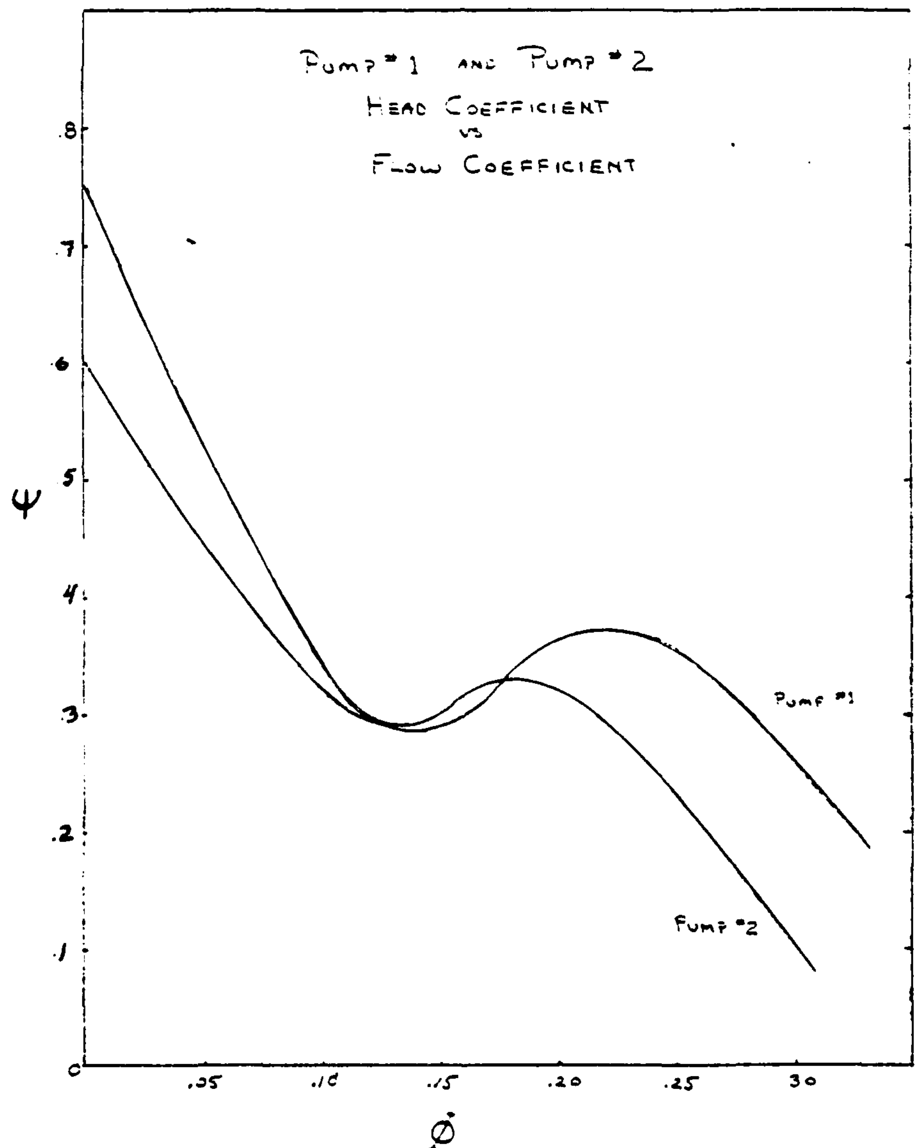


FIGURE 45.

Pump #1 And Pump #2 Characteristics



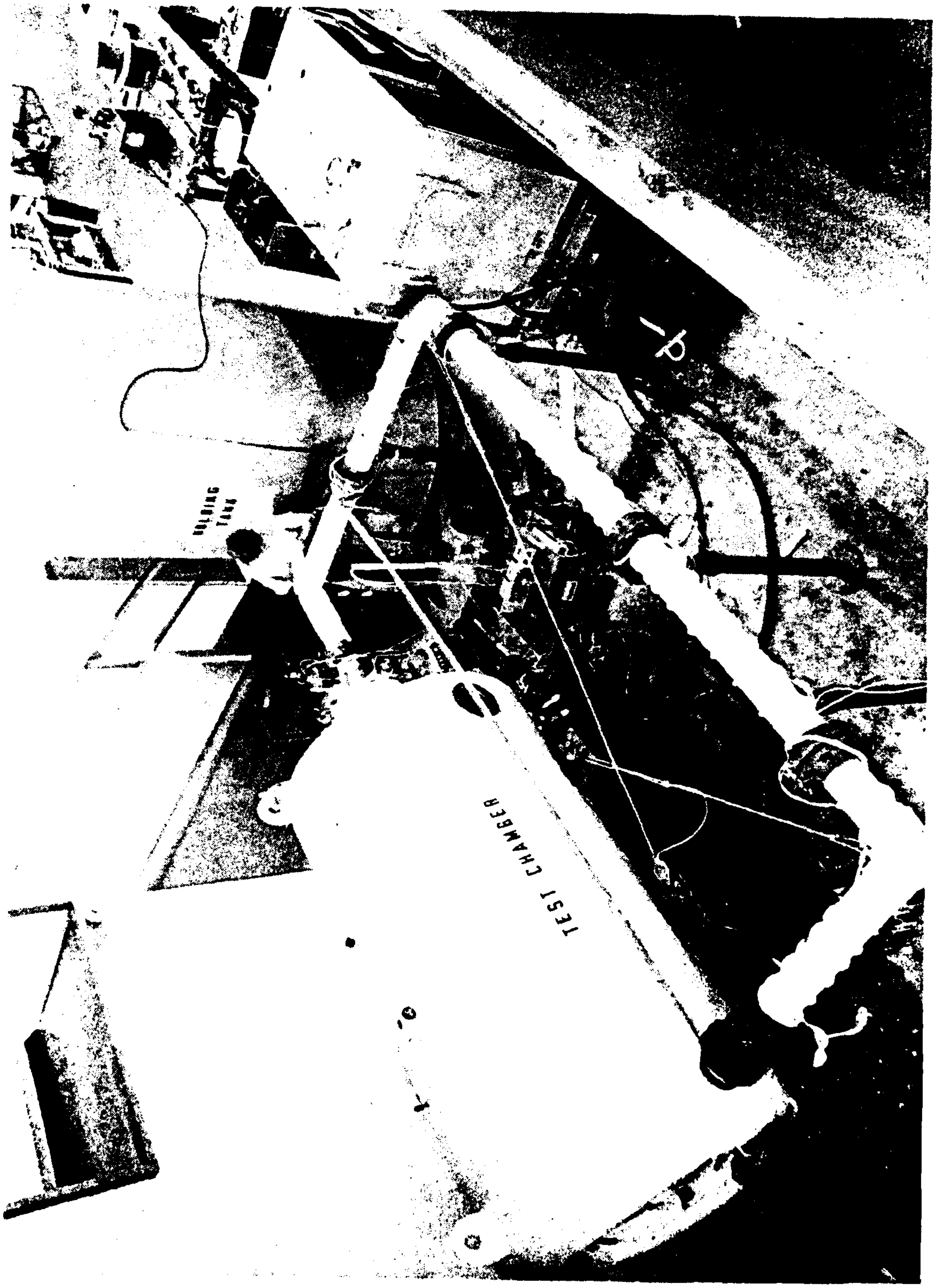


FIGURE 46. Flow Loop and Test Chamber

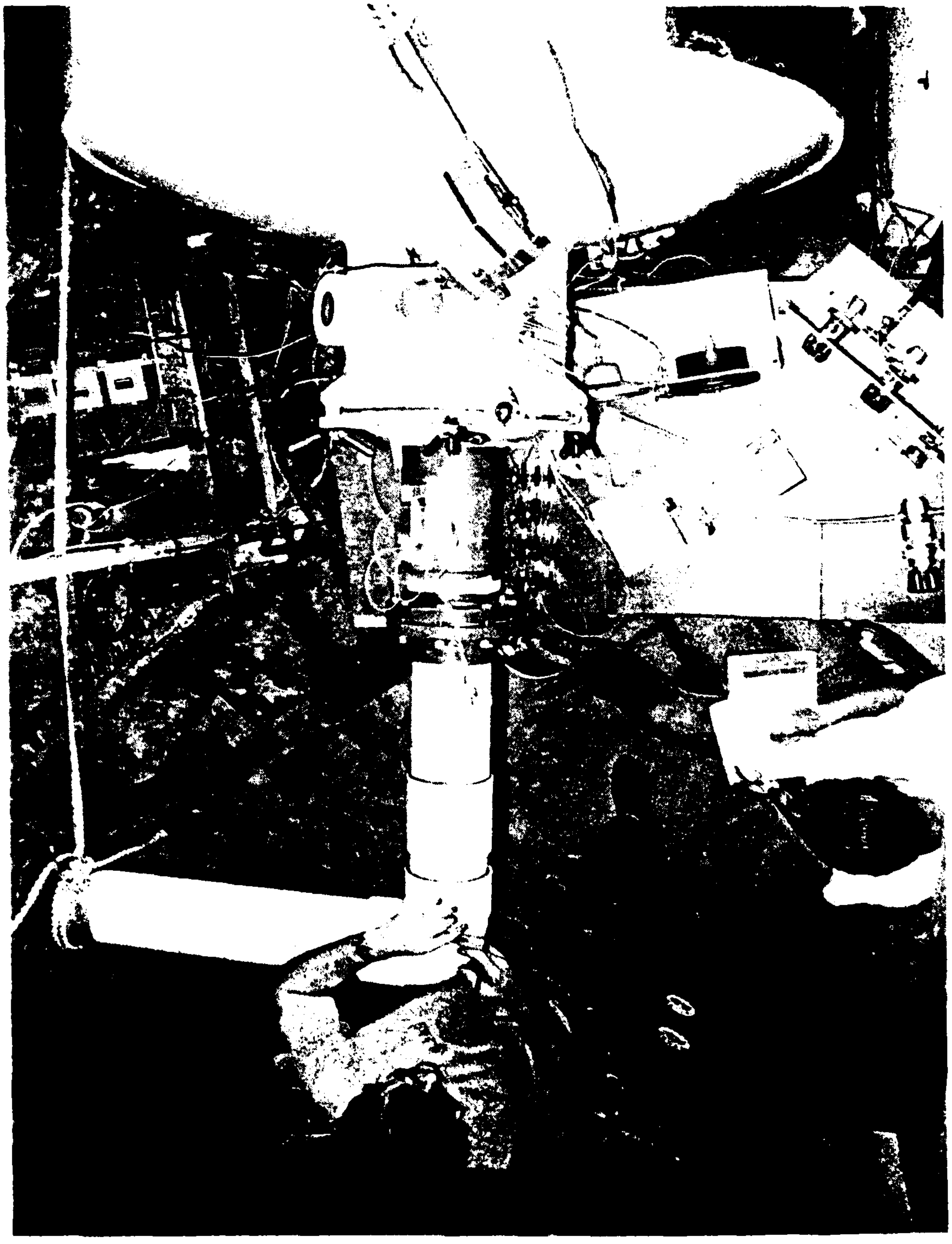


FIGURE 47. Turbine Installation and Instrumentation

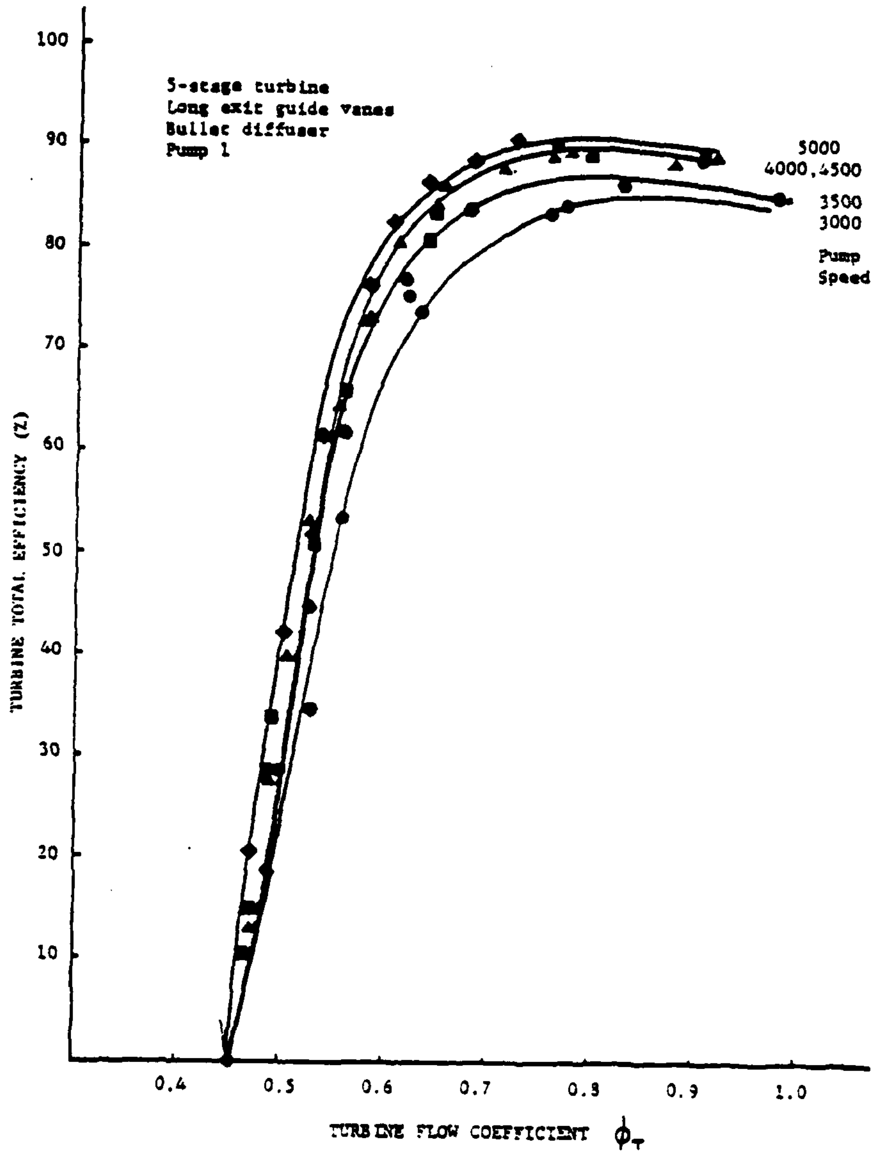


FIGURE 48.

Turbine Efficiency
Versus
Flow Coefficient

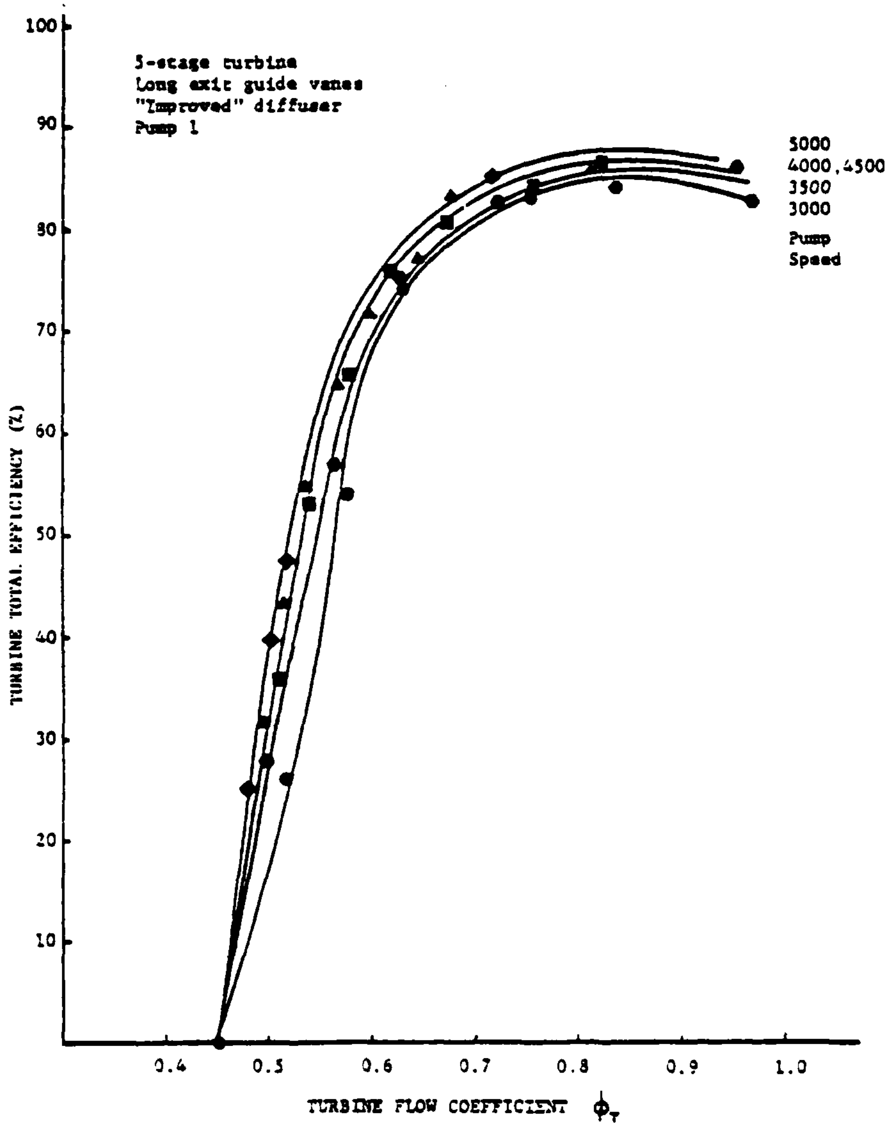


FIGURE 49.

Turbine Efficiency
Versus
Flow Coefficient

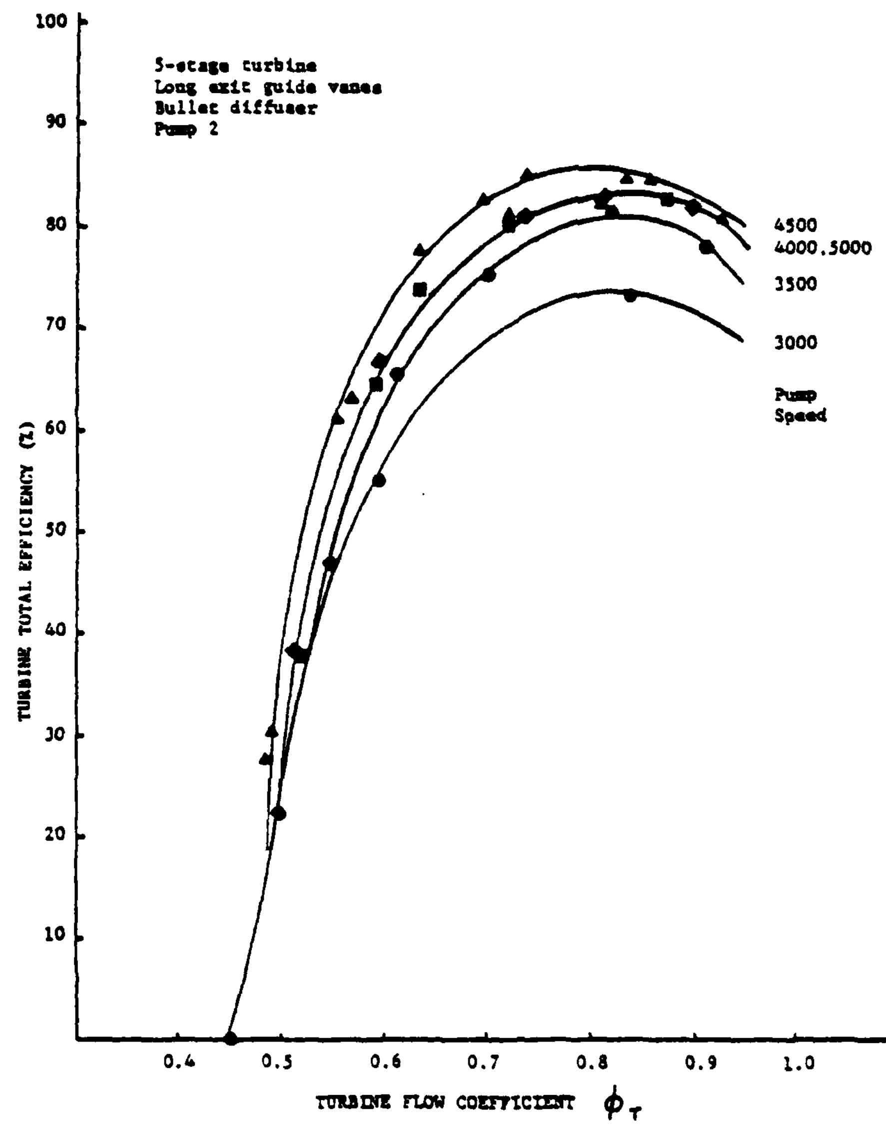


FIGURE 50.

Turbine Efficiency
Versus
Flow Coefficient

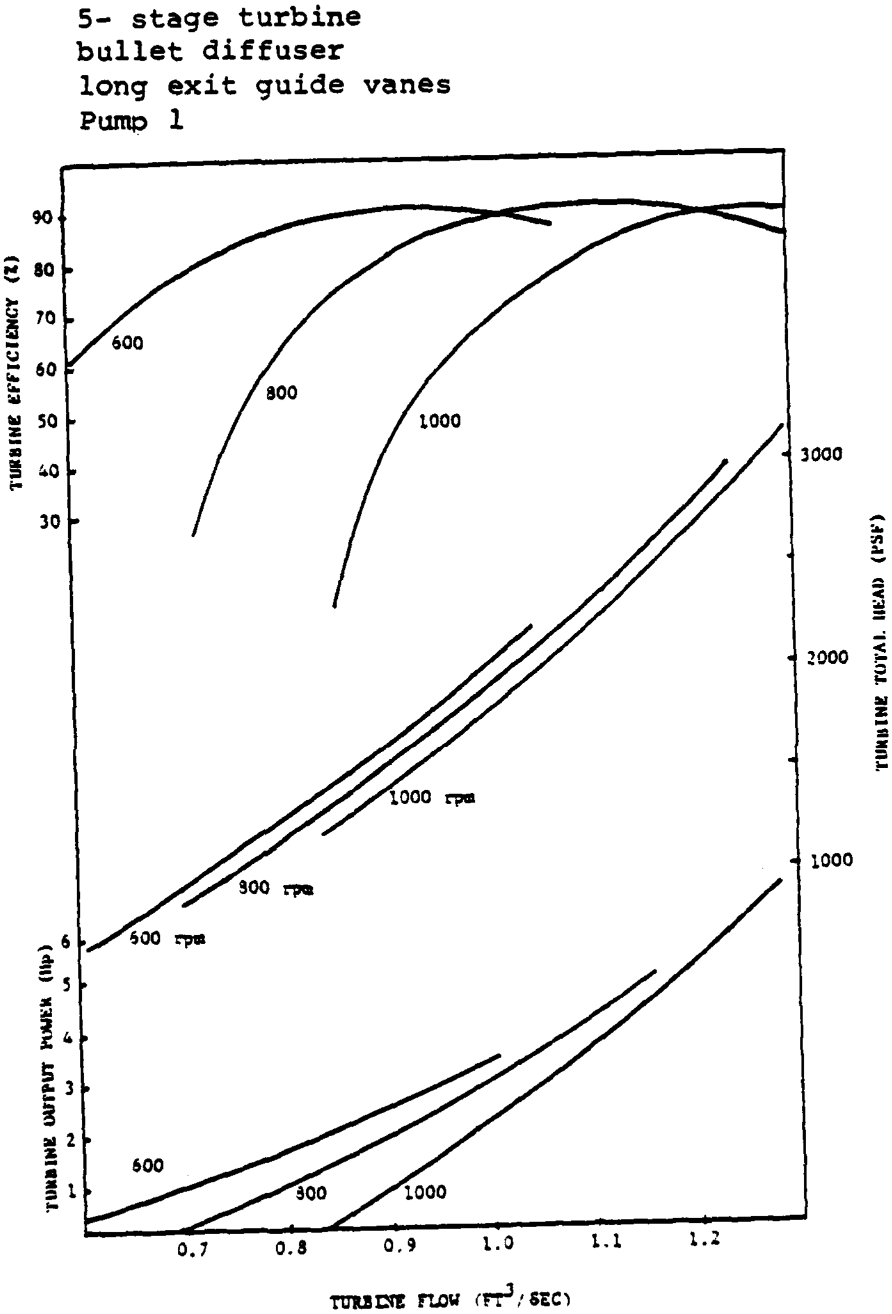


FIGURE 51.

Turbine Performance

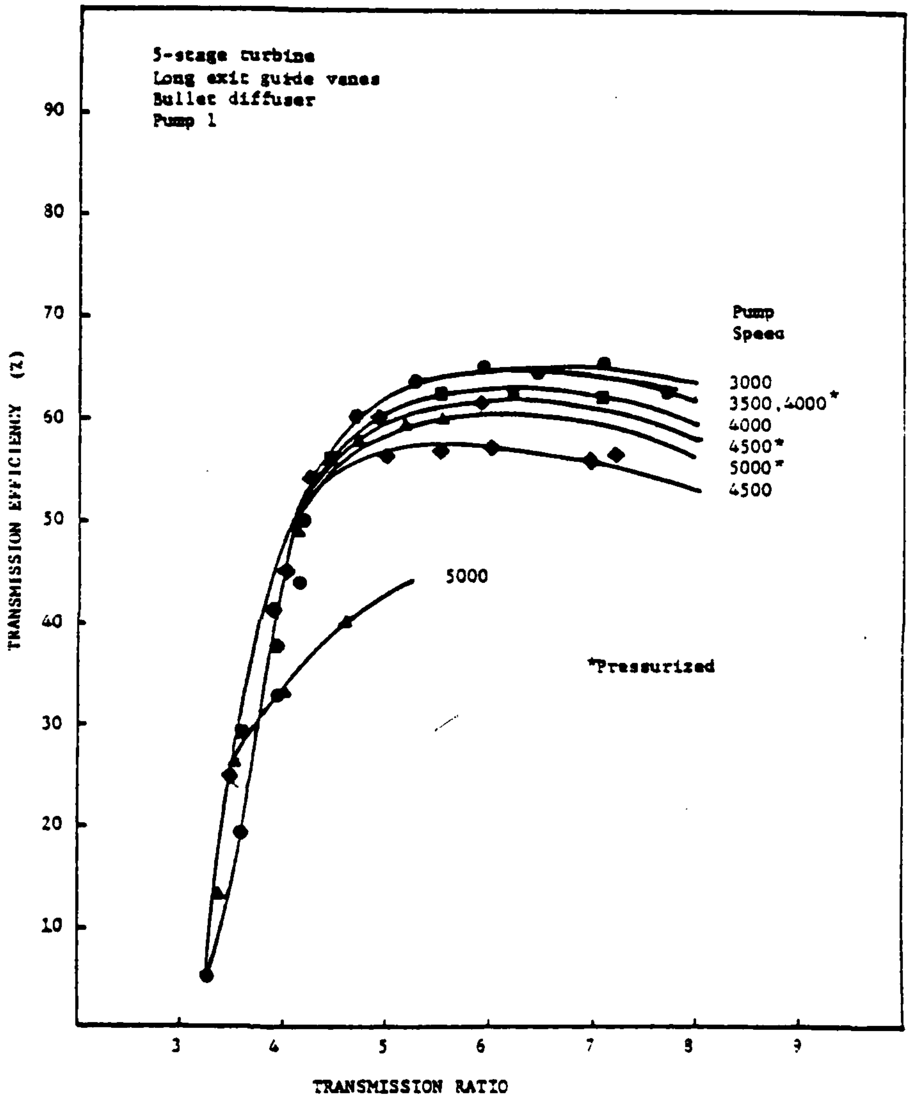


FIGURE 52.

Transmission Efficiency
Versus
Transmission Ratio

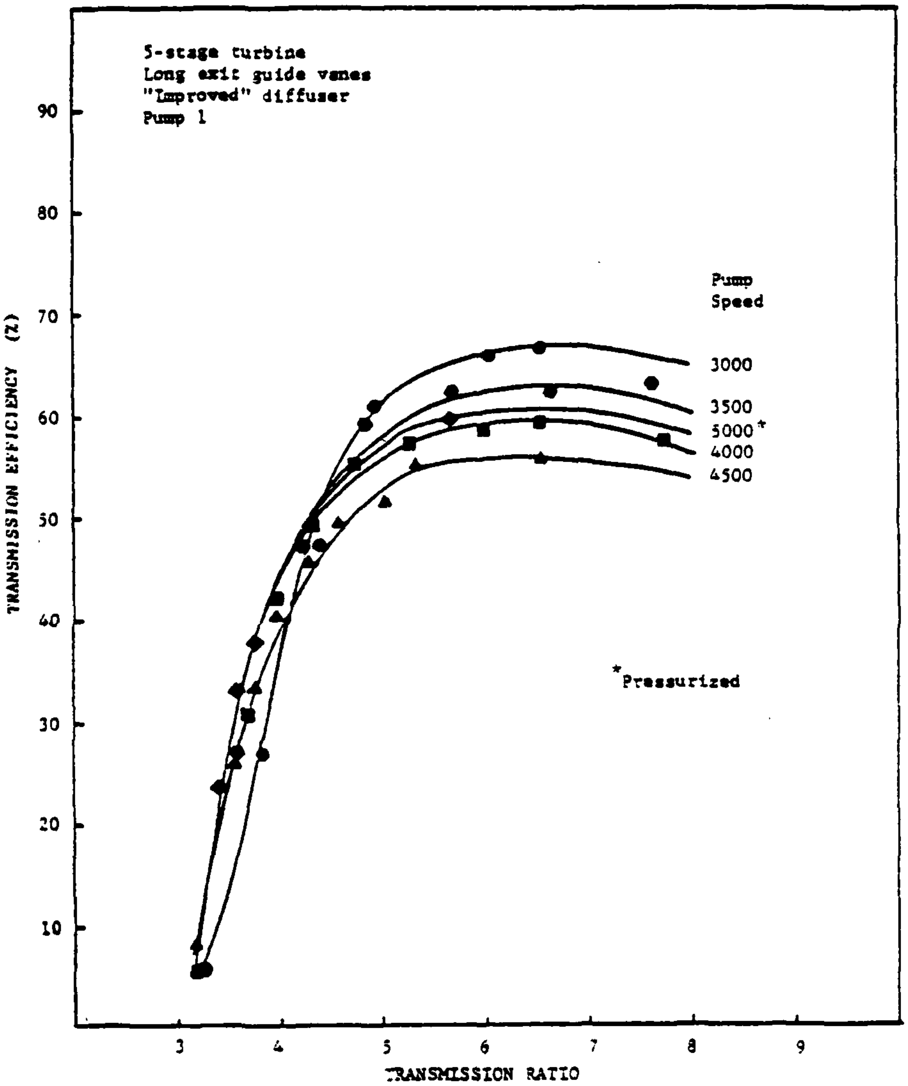


FIGURE 53.

Transmission Efficiency
Versus
Transmission Ratio

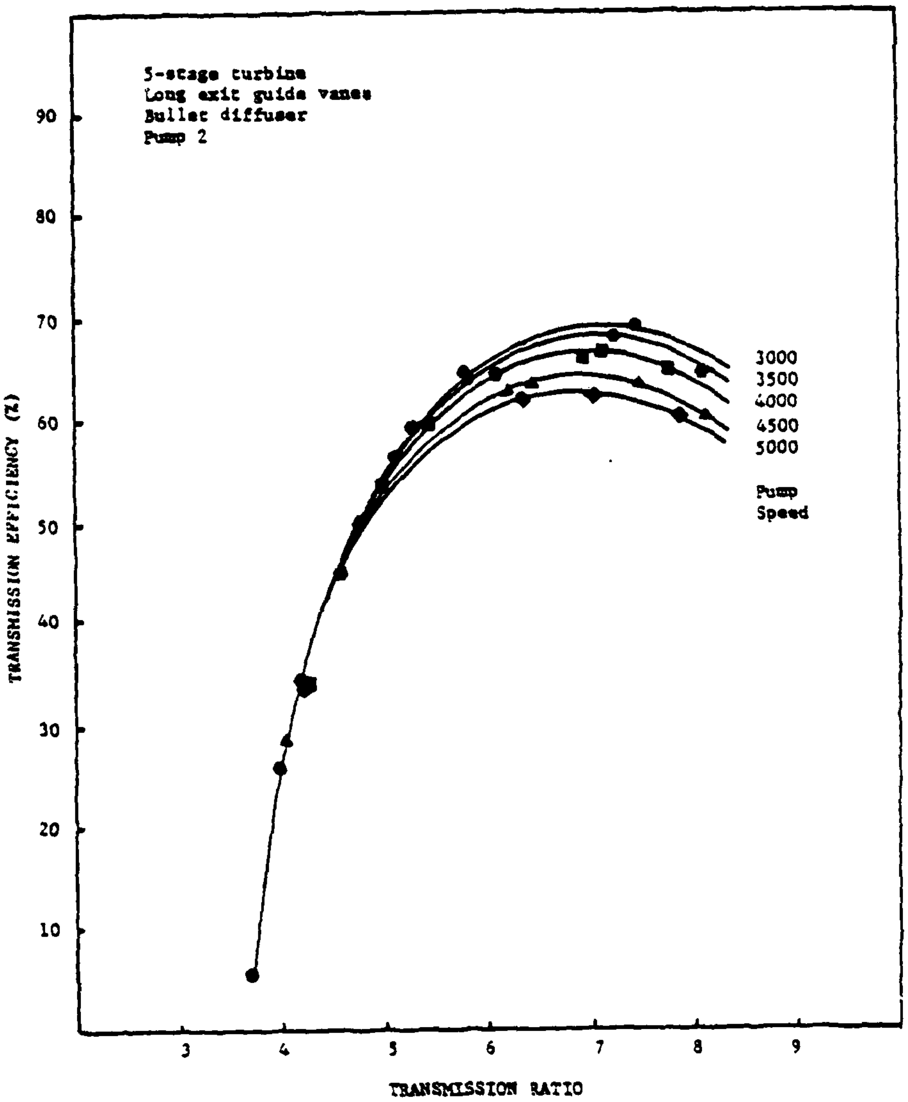


FIGURE 54.

Transmission Efficiency
Versus
Transmission Ratio

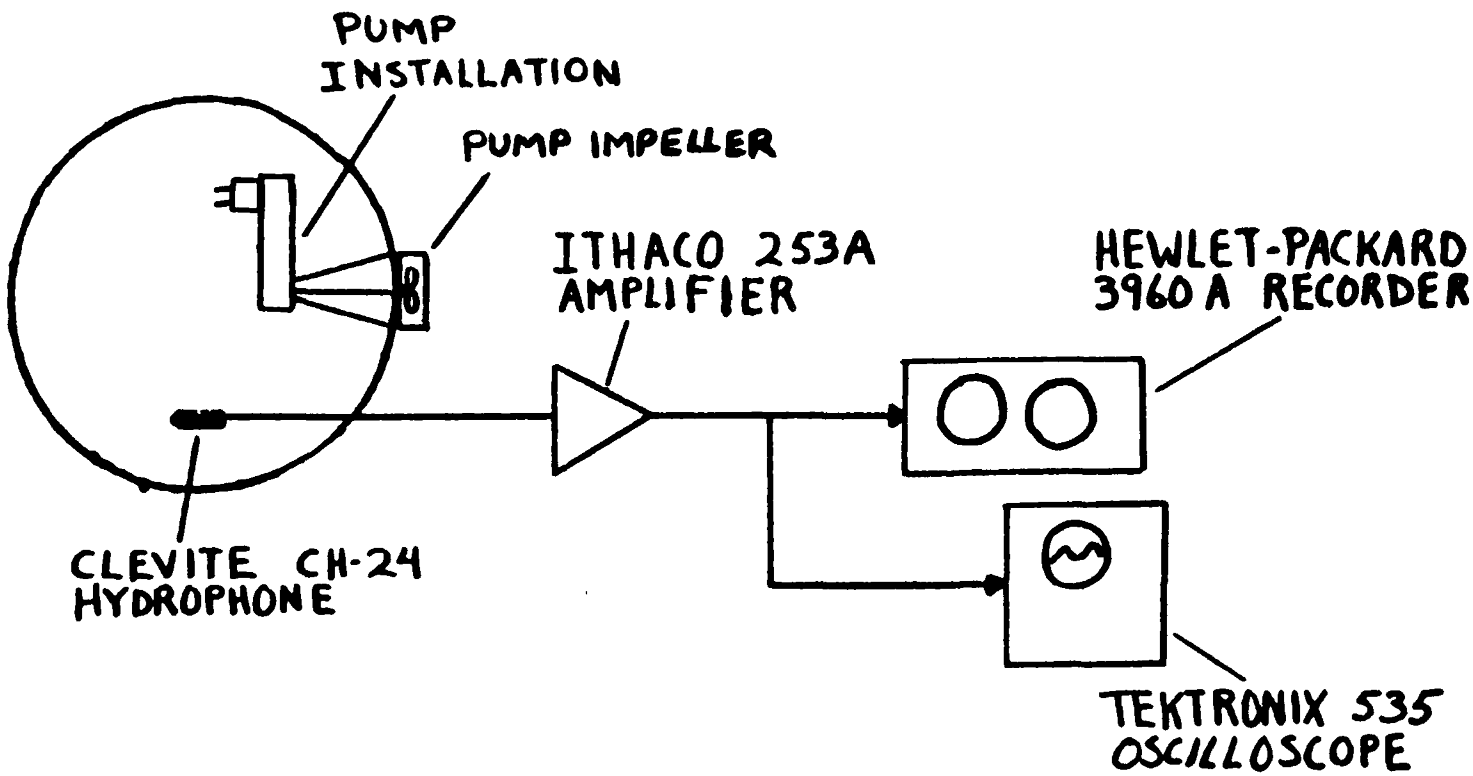


FIGURE 55.

MEASUREMENT SYSTEM

AD-A083 932 RHODE ISLAND UNIV KINGSTON DEPT OF OCEAN ENGINEERING F/G 13/10
A NEW PROPULSION SYSTEM FOR SHIPS.(U)
JAN 80 H E SHEETS, T KOWALSKI, A P DAVIS N00014-76-C-0101

UNCLASSIFIED NL

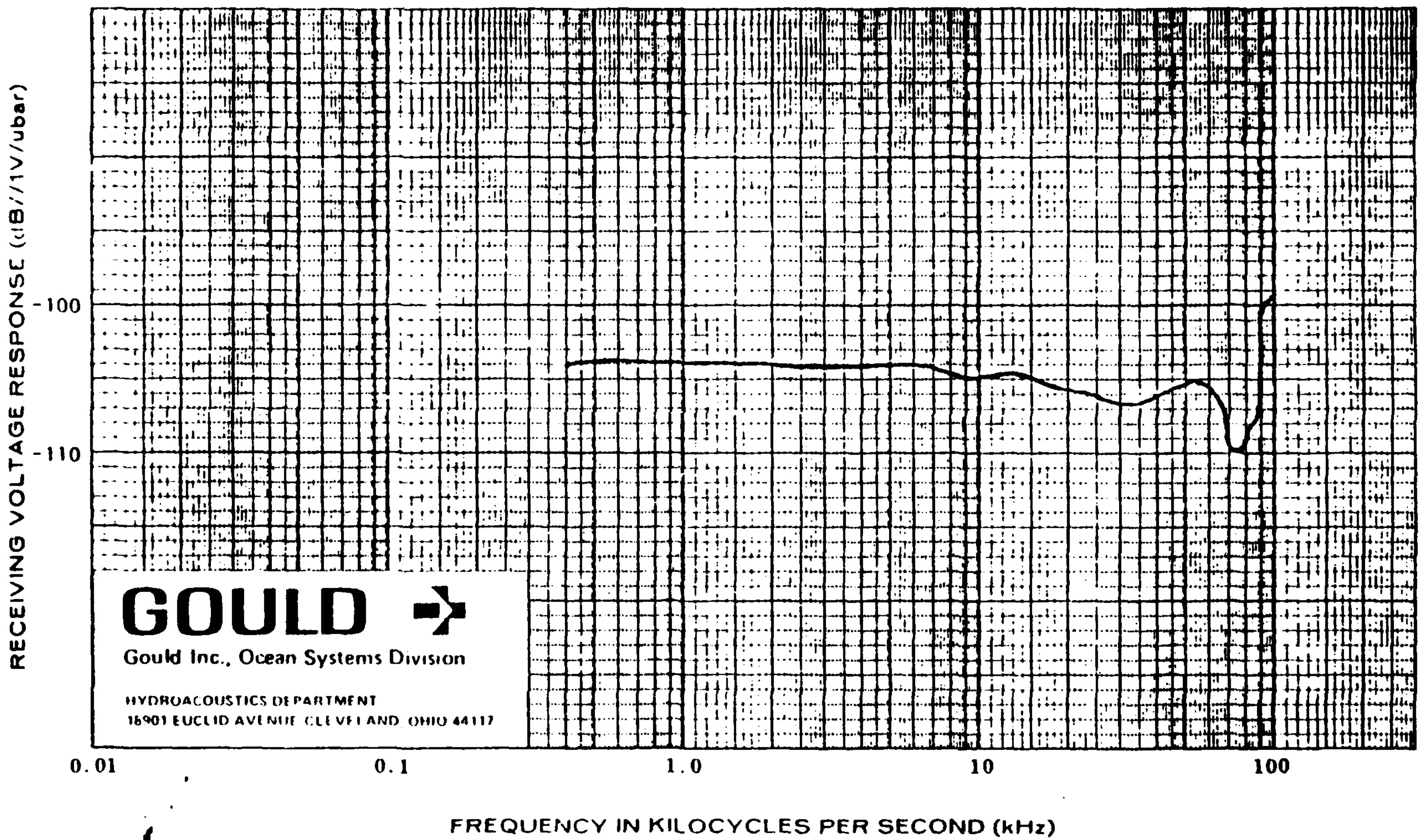
3 of 3

AD
A083932

END
DATE
FILMED
6 80
DTIC

FIGURE 56.

HYDROPHONE SENSITIVITY CALIBRATION



Model: CH-24 Serial Number: 1 Water Temperature: 5°C Distance: 1 M Depth: 7.6 M

Method: Substitution Date: 6 Feb. 1978 Reference Standard: CH1A S/N 2009

☐ Sensor Terminals ☒ End of Cable: Reference Standard Calibration Date: December 1977

MEASUREMENTS MADE IN ACCORDANCE WITH ASA STANDARD Z 24 24 1957

CO 94b 4 77

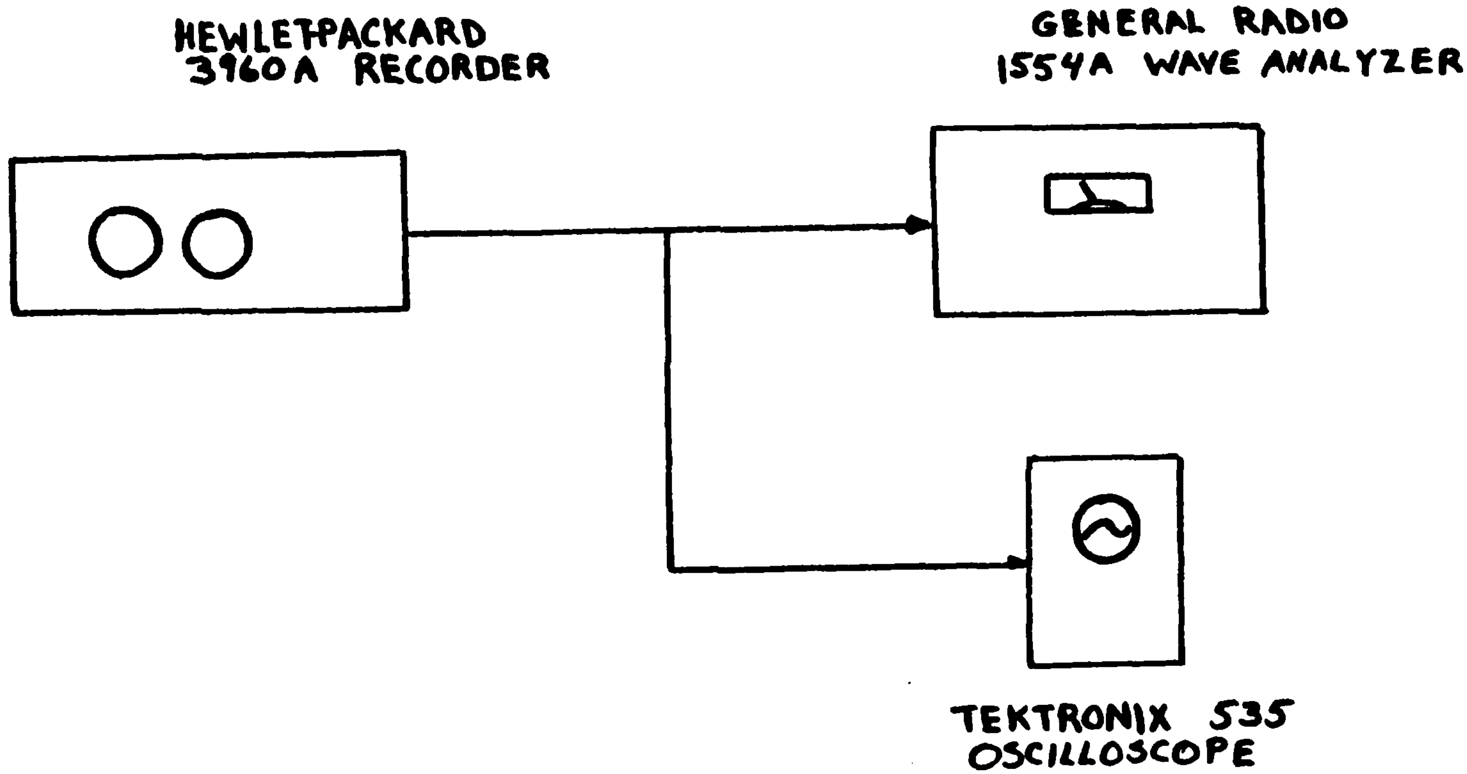


FIGURE 57.

FREQUENCY ANALYSIS SYSTEM

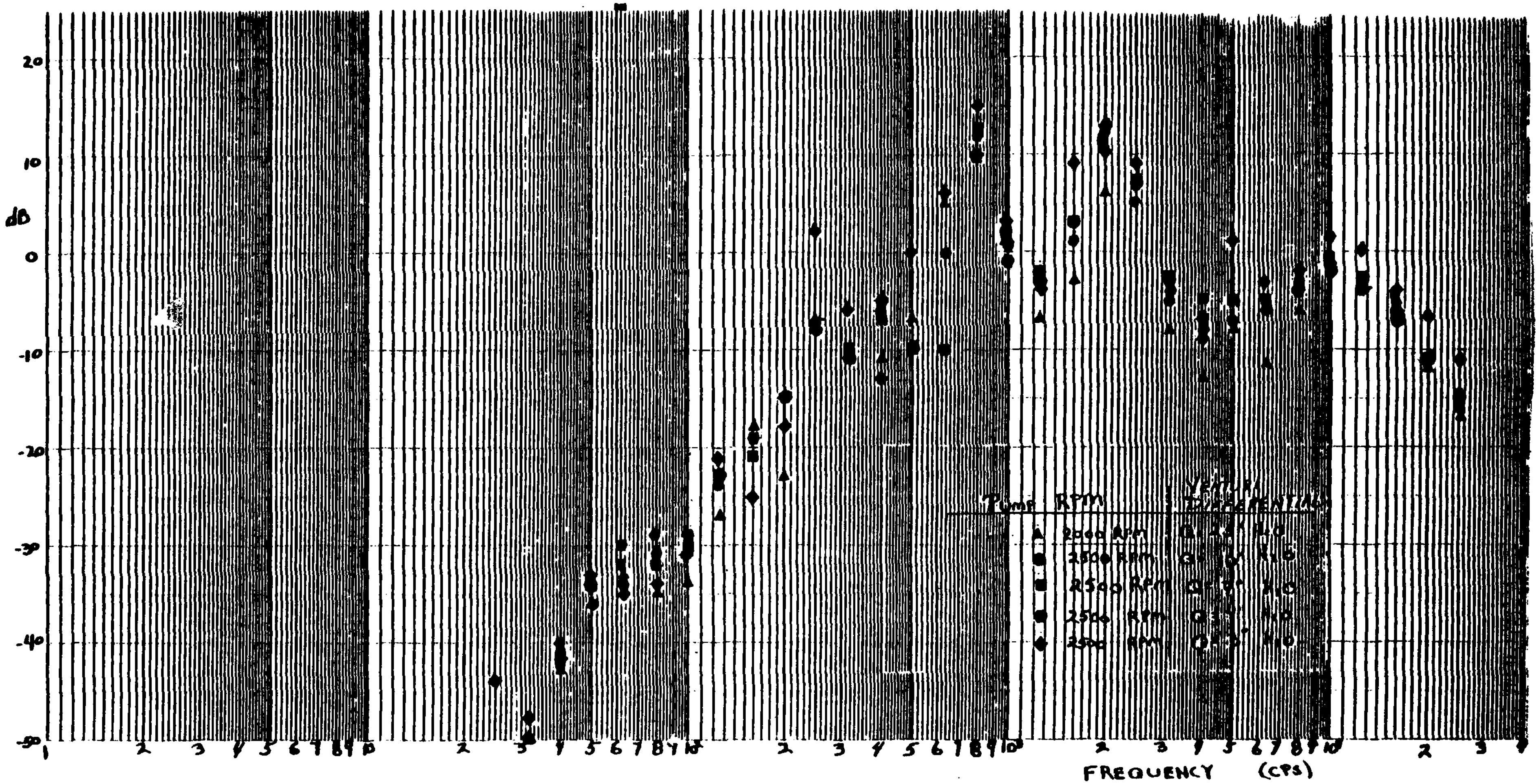


FIGURE 58.

Noise Power Versus Frequency At 2500 RPM

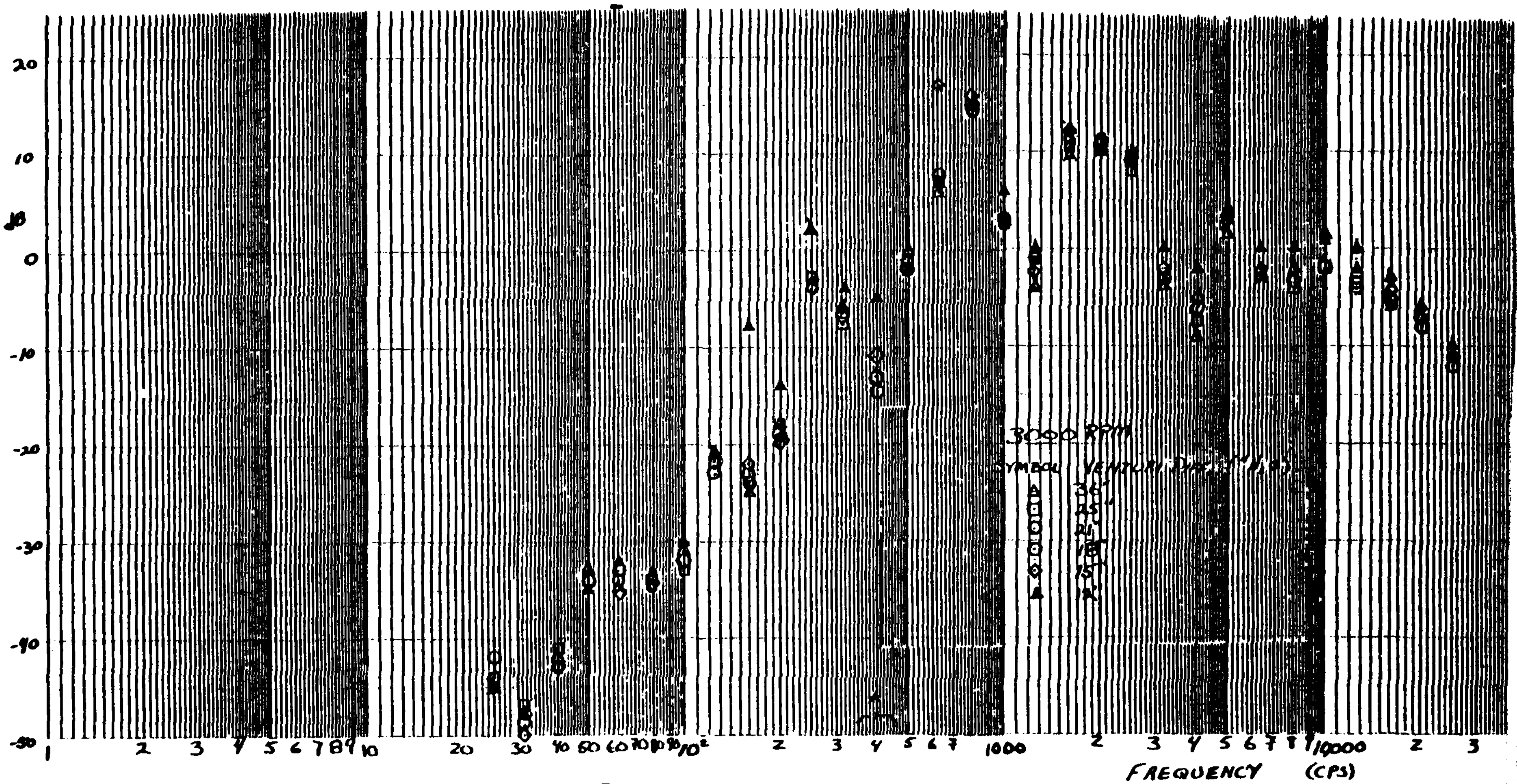


FIGURE 59.

Noise Power Versus Frequency At 3000 RPM

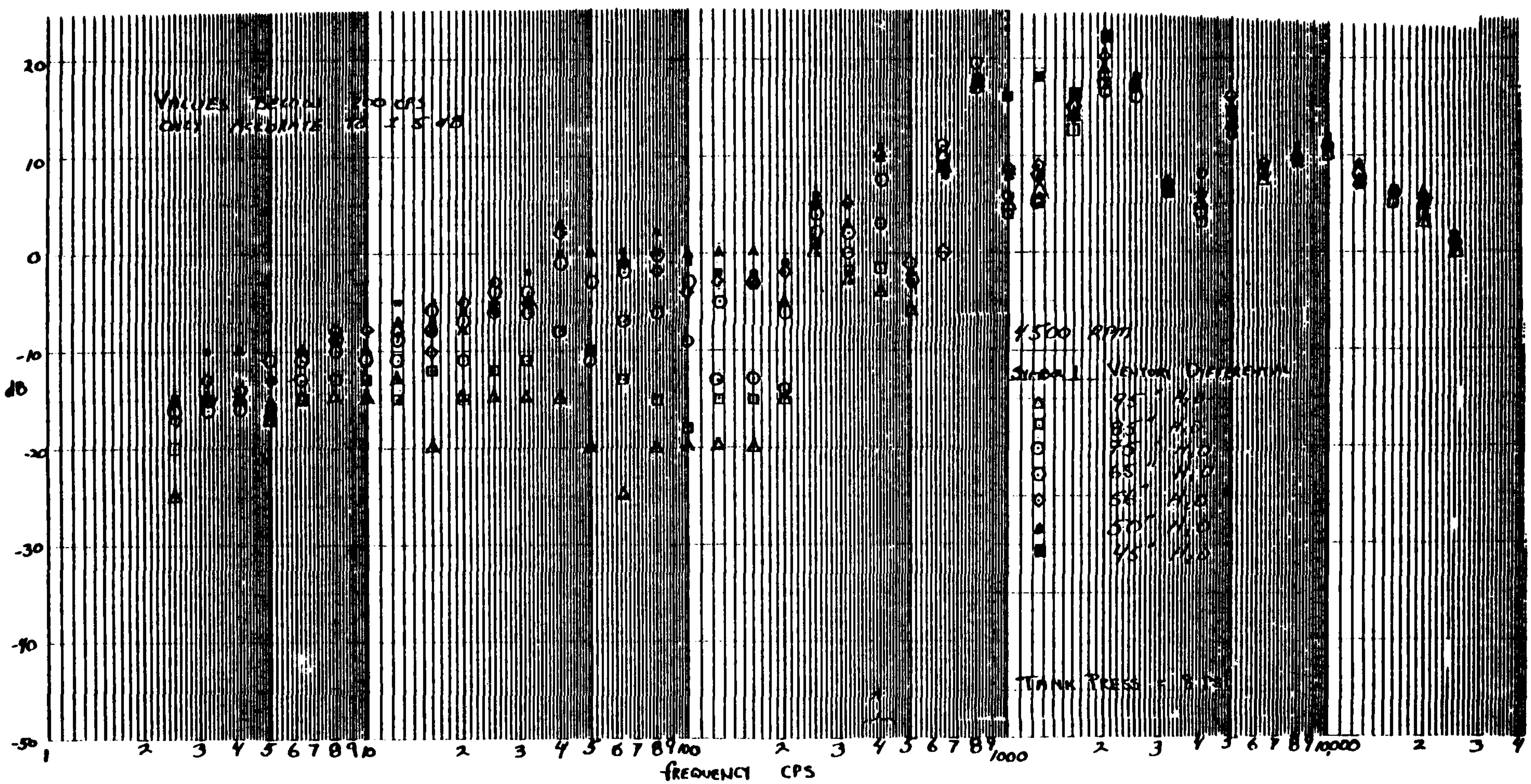


FIGURE 60.

Noise Power Versus Frequency At 4500 RPM

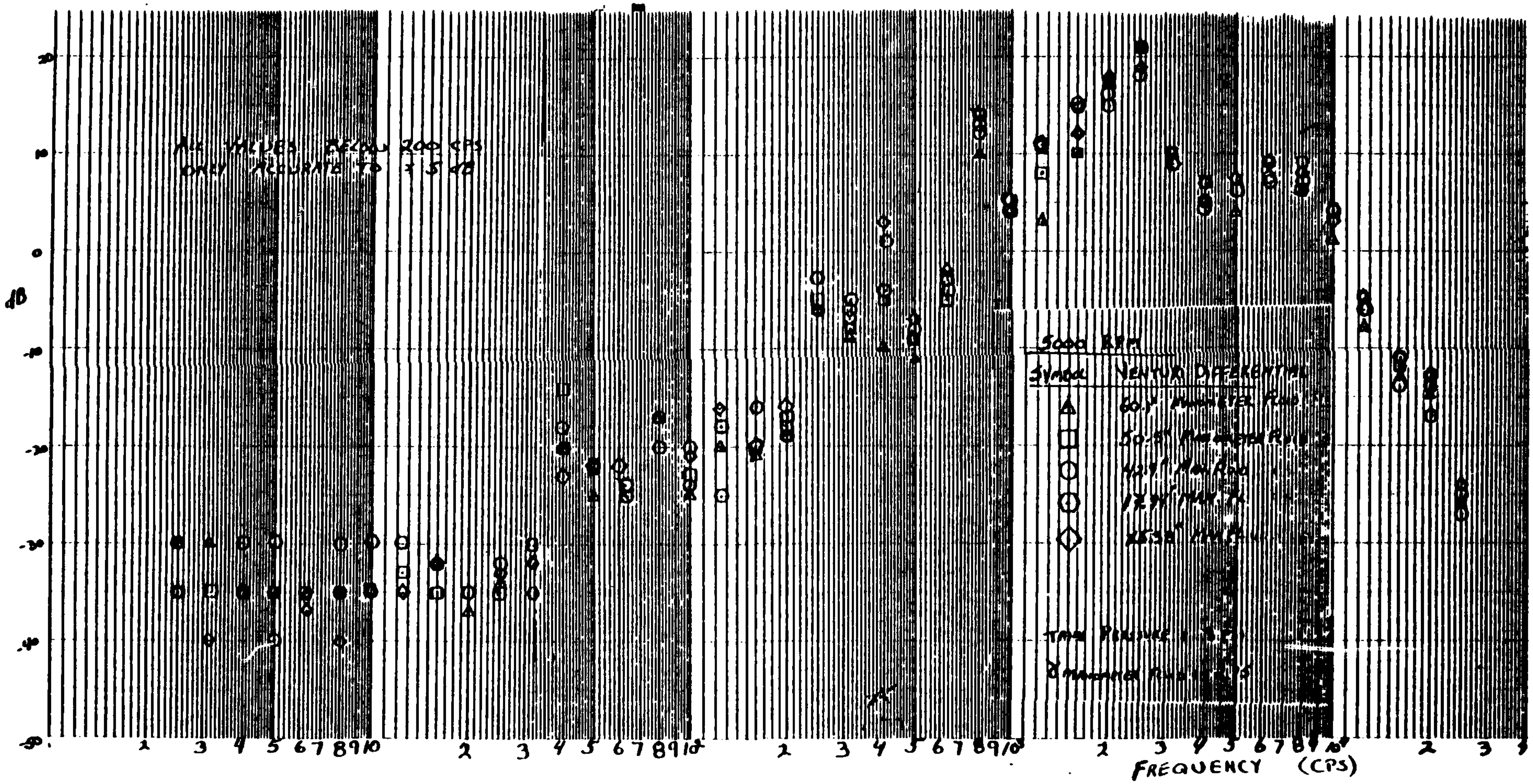


FIGURE 61.

Noise Power Versus Frequency At 5000 RPM

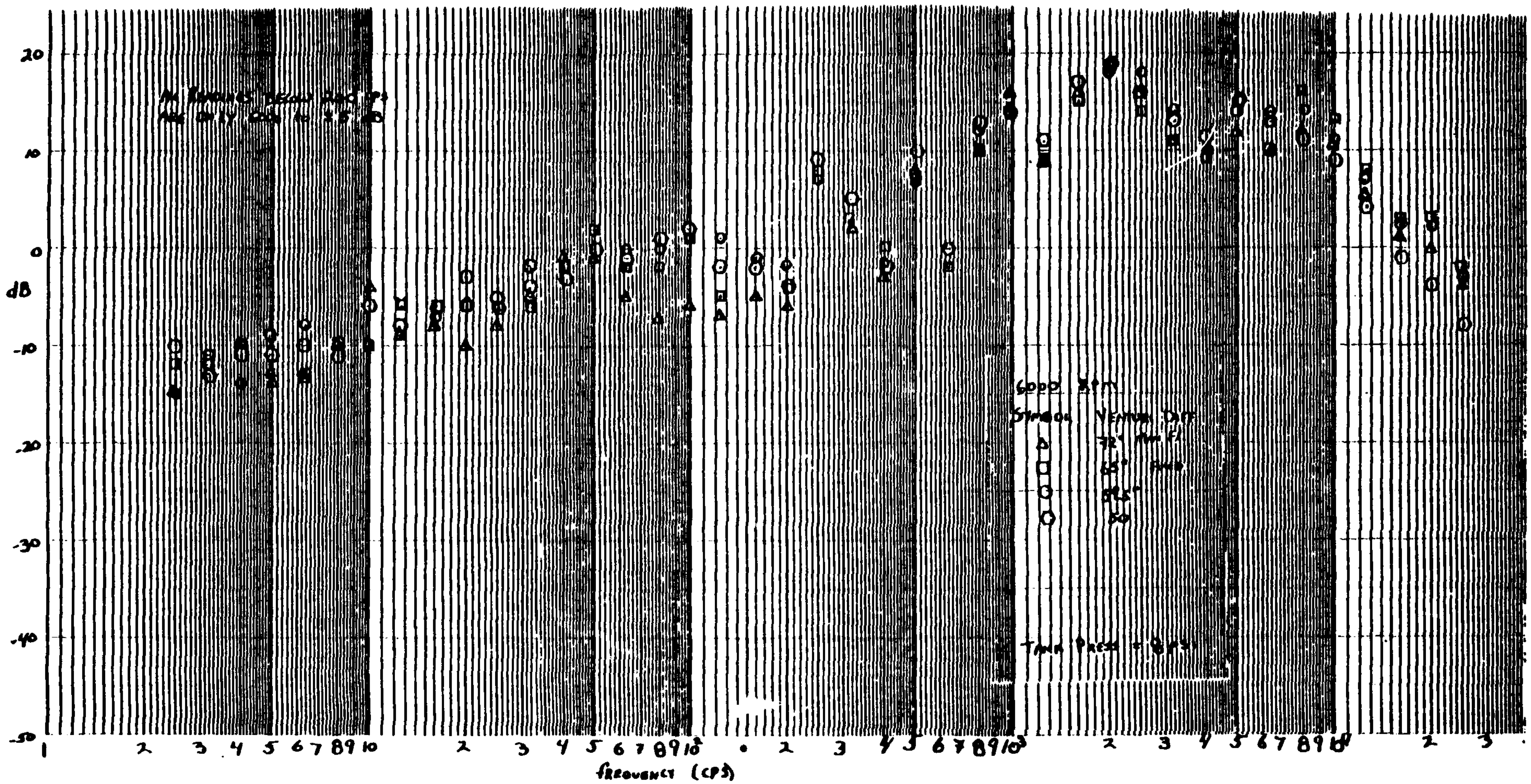


FIGURE 62.

Noise Power Versus Frequency At 6000 RPM

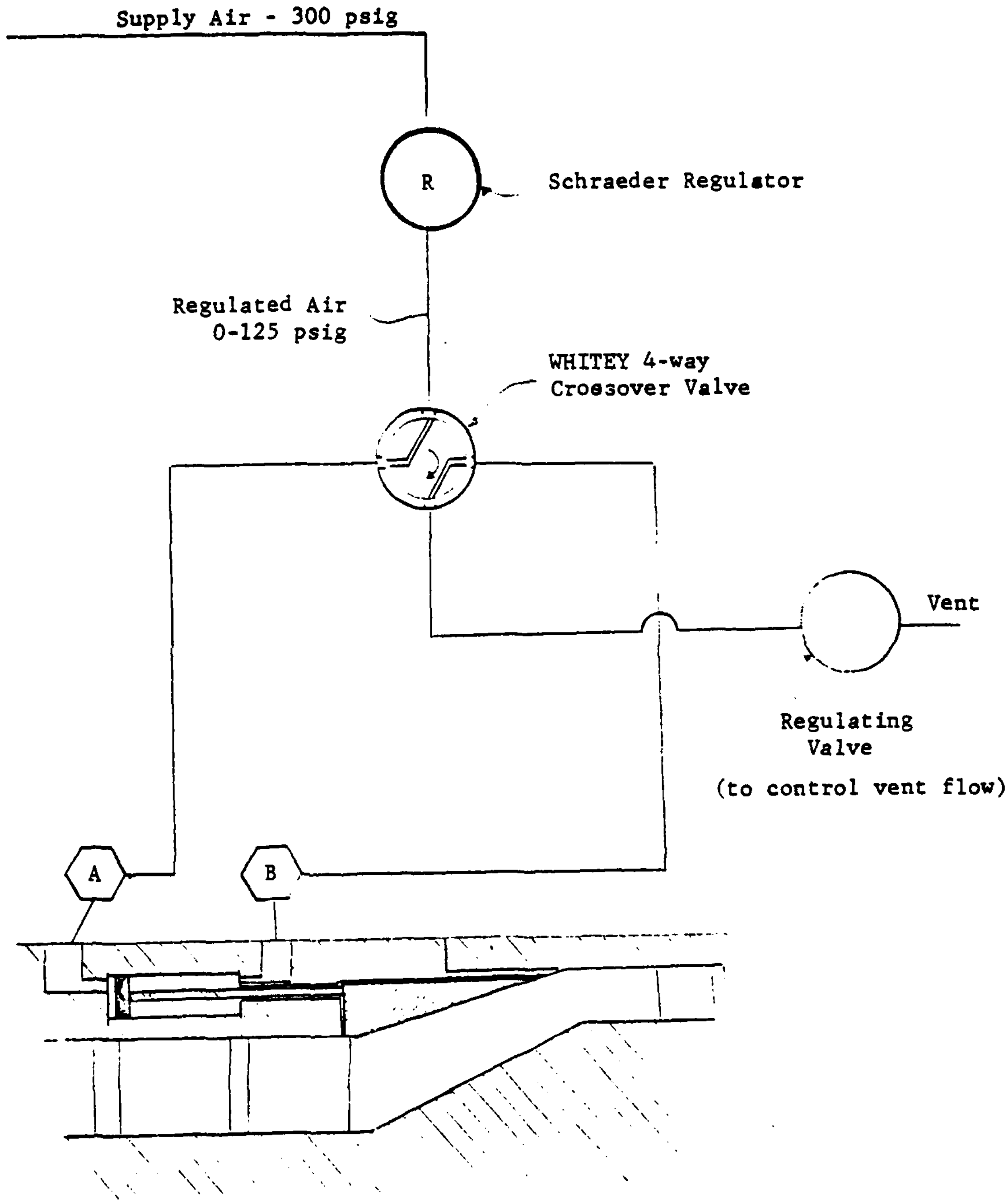
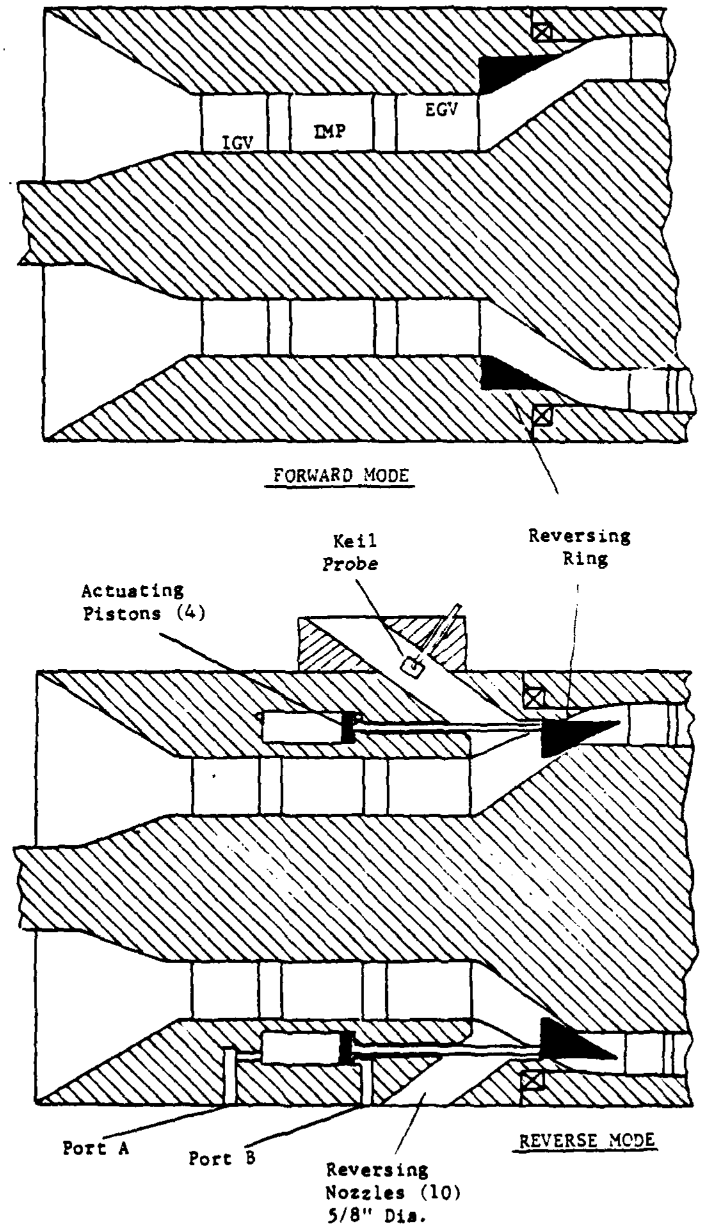


FIGURE 63.

Reversing Mechanism And Actuation System

FIGURE 64.

Schematic of Reversing Mechanism



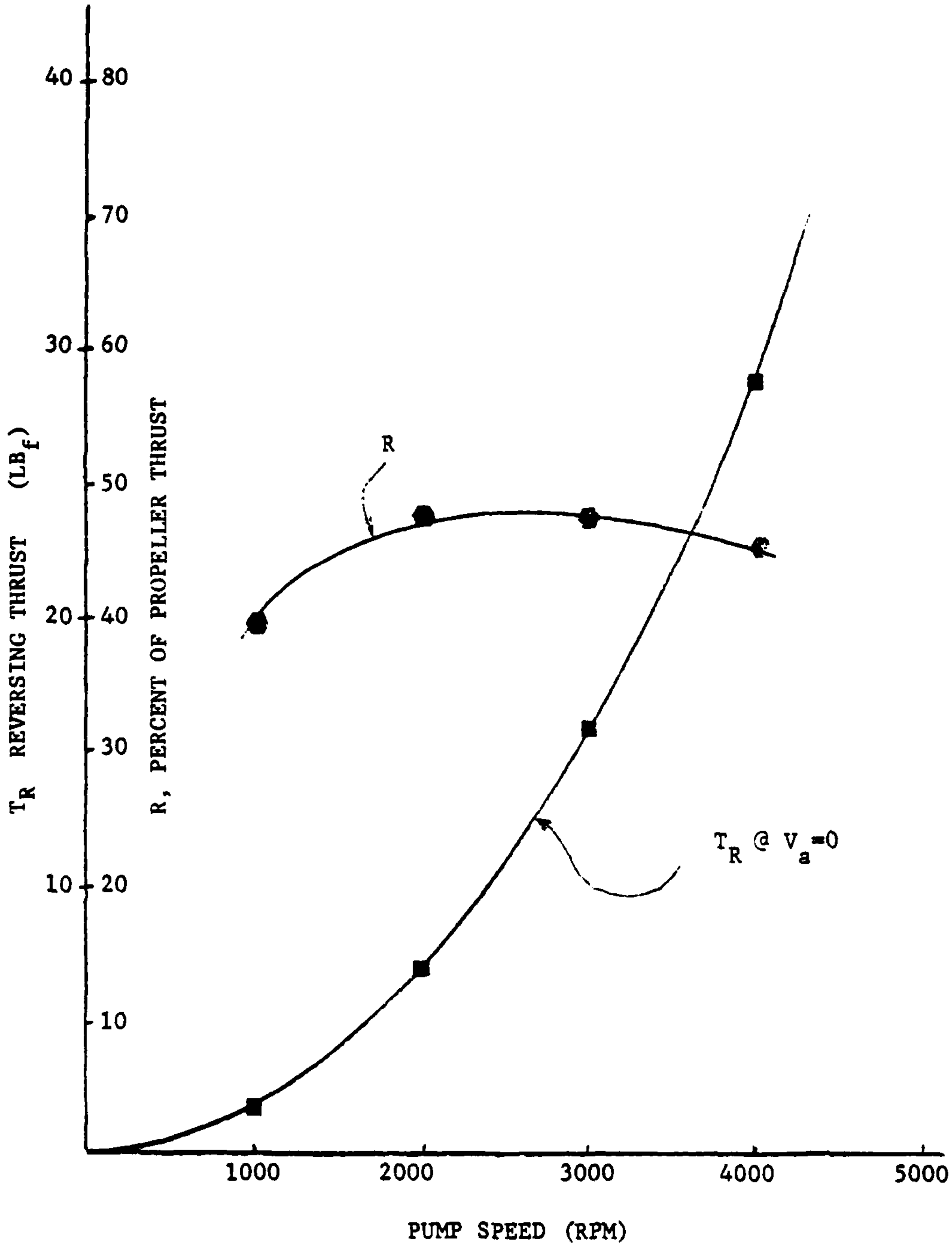


FIGURE 65.

Performance of Reversing System

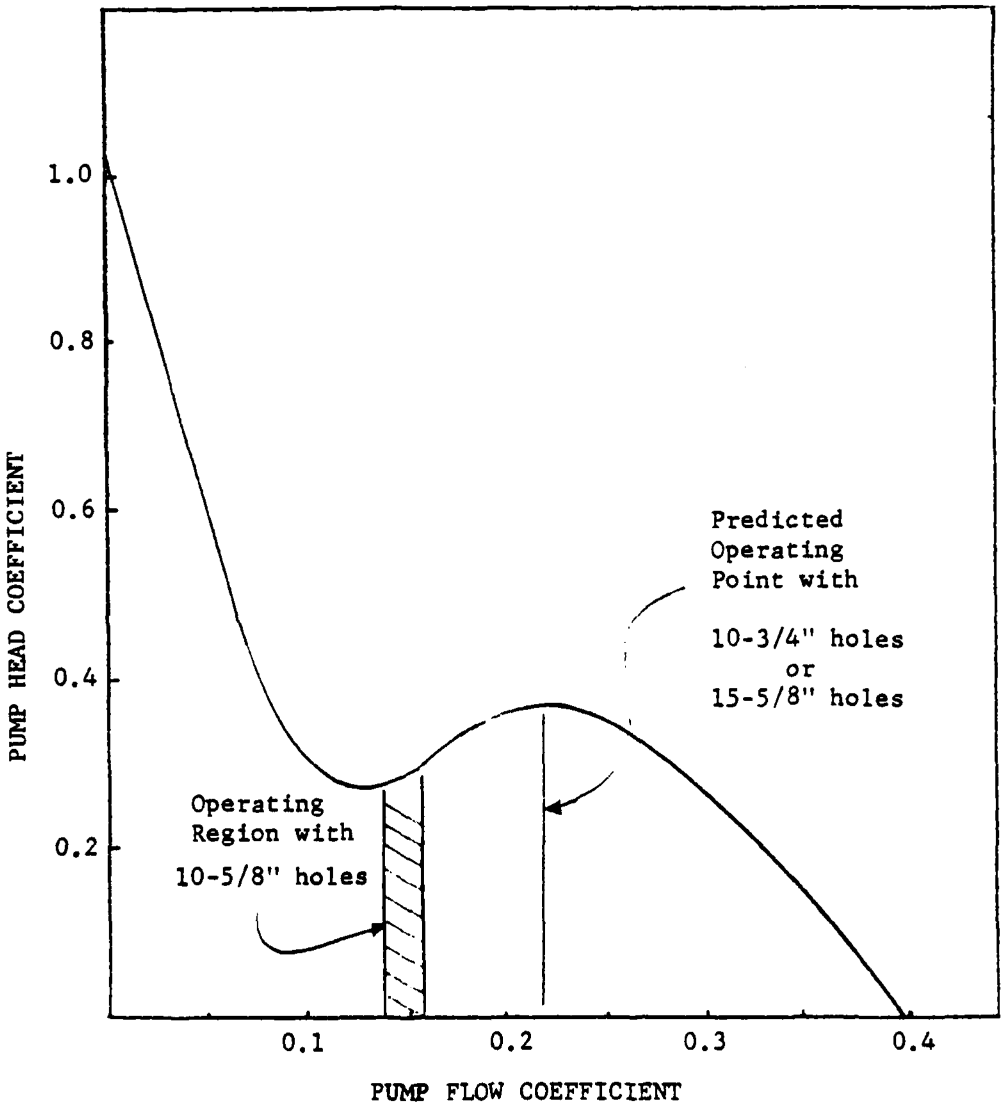


FIGURE 66.

Pump Characteristics And Reversing System

FIGURE 67.

TEST INSTALLATION--
REVERSE THRUST MODE FLOW PATH

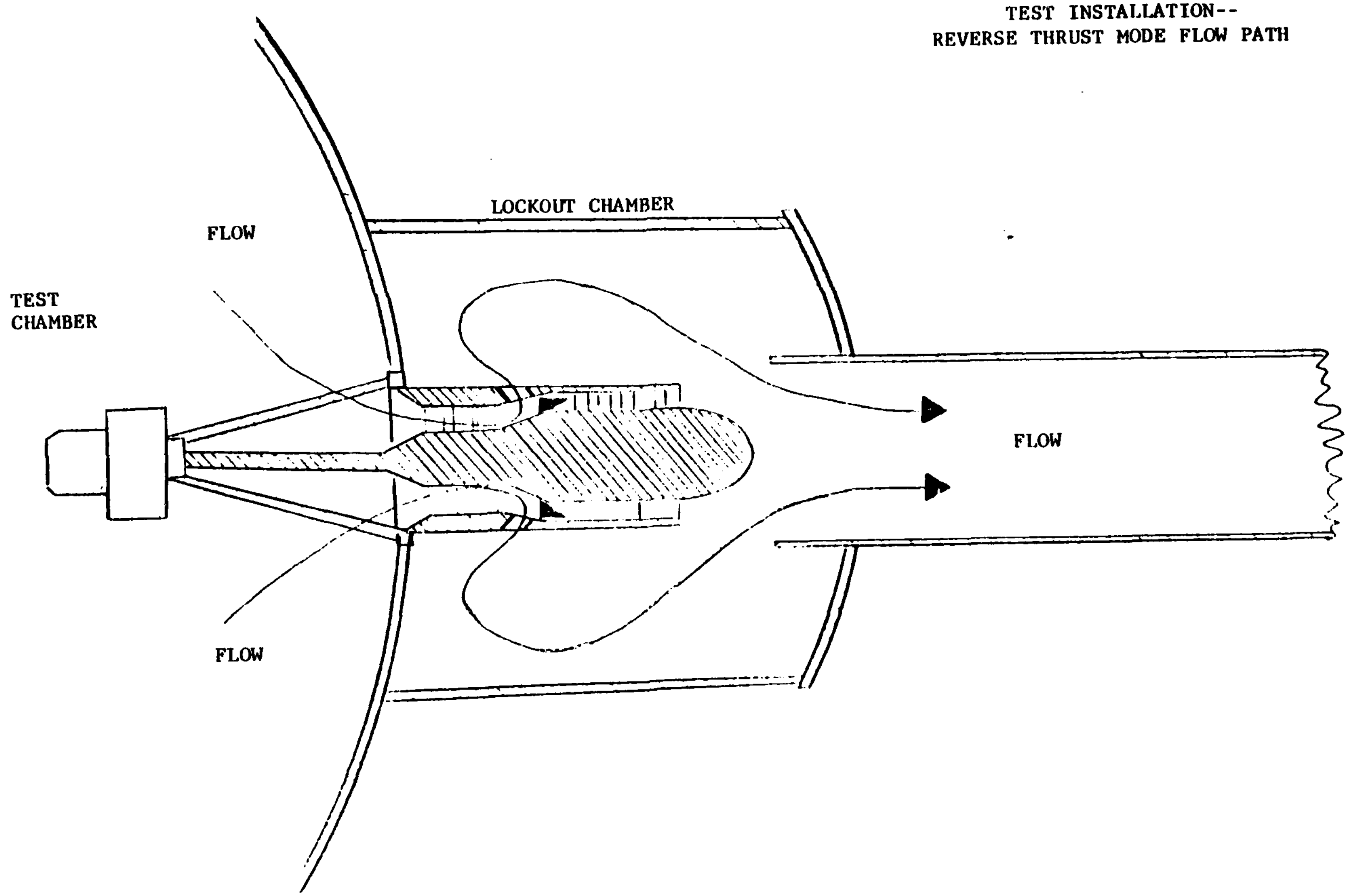
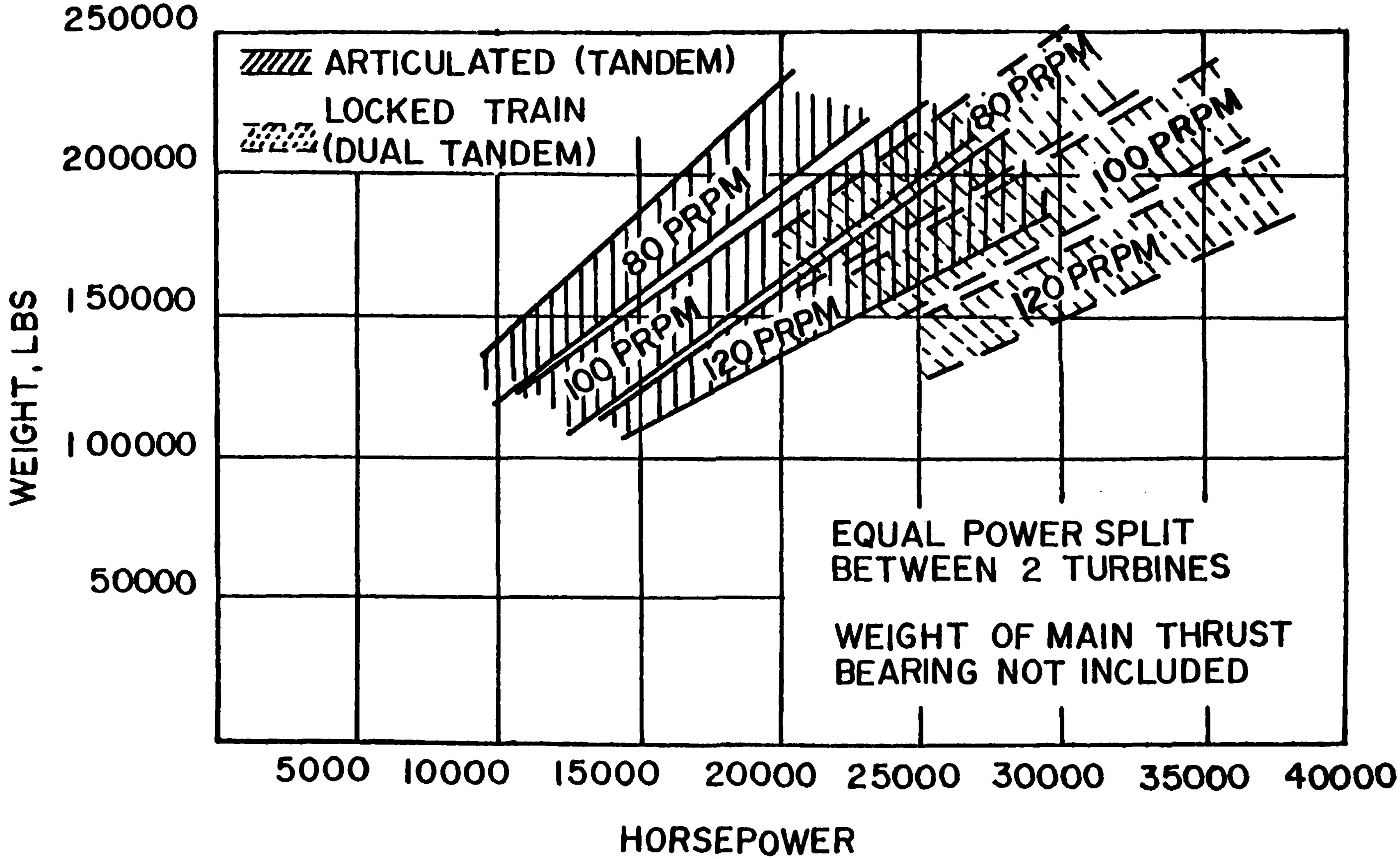
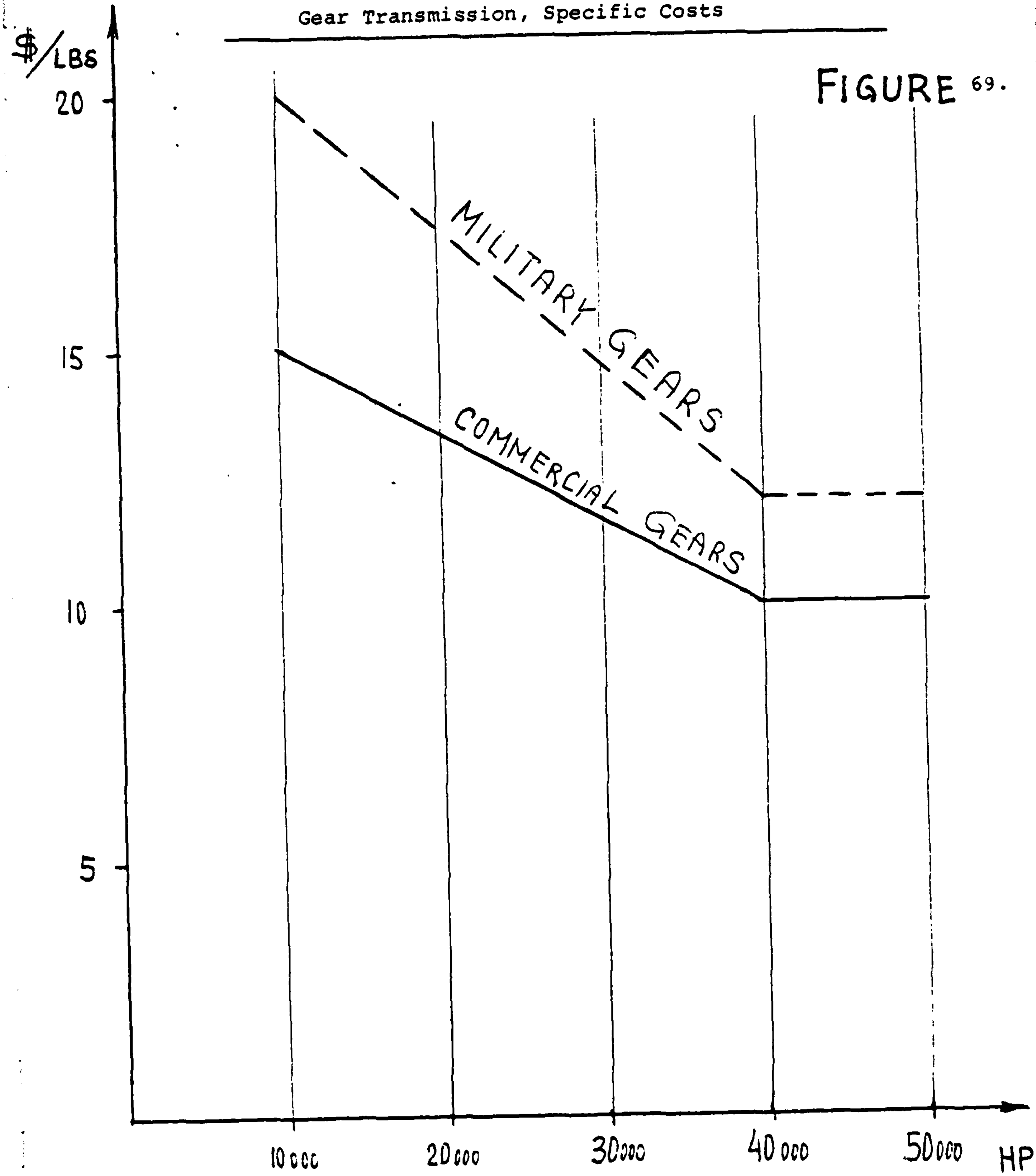


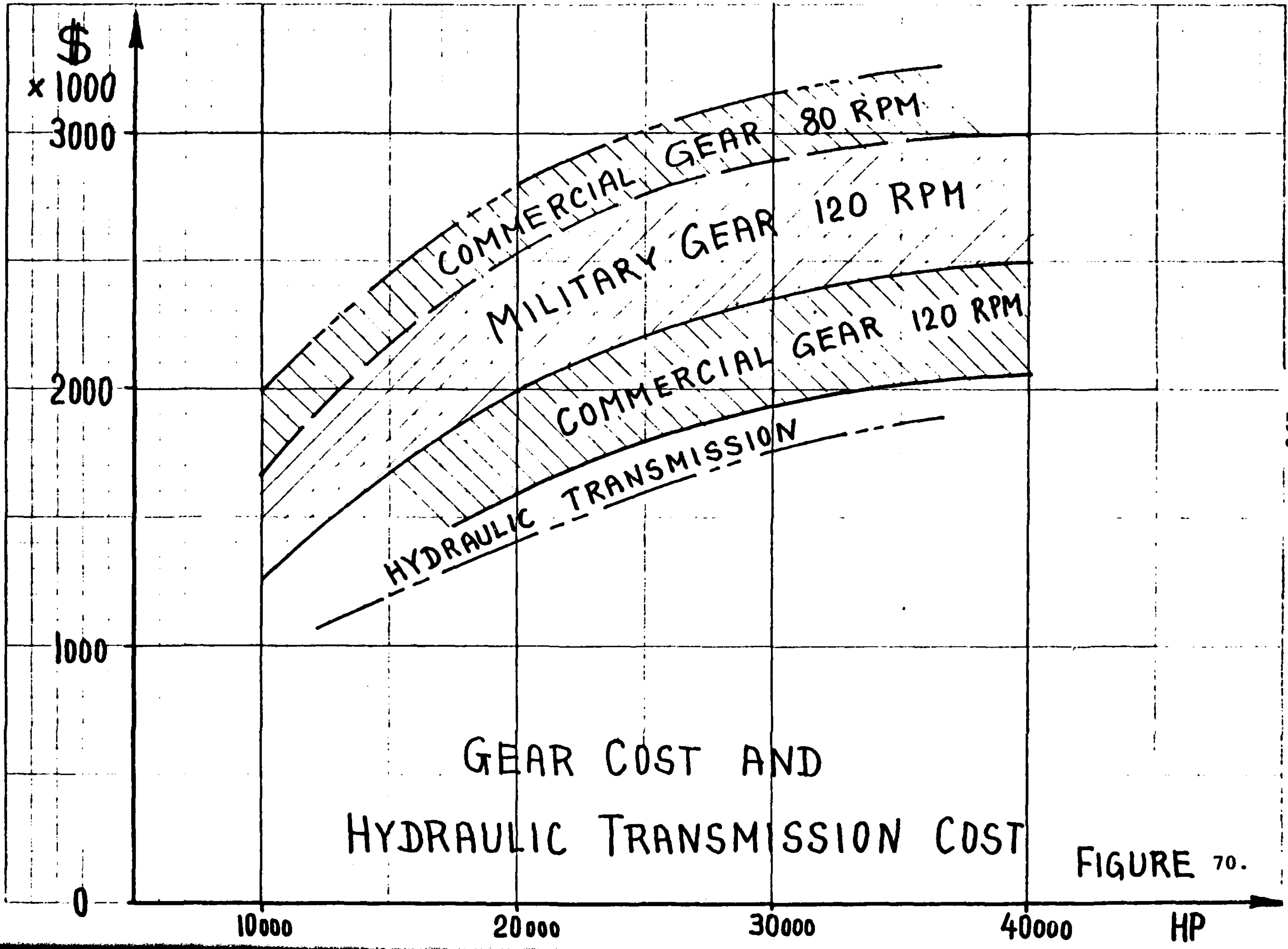
FIGURE 68. Gear Transmission, Weight Versus Horsepower



Gear Transmission, Specific Costs

FIGURE 69.





GEAR COST AND
HYDRAULIC TRANSMISSION COST

FIGURE 70.

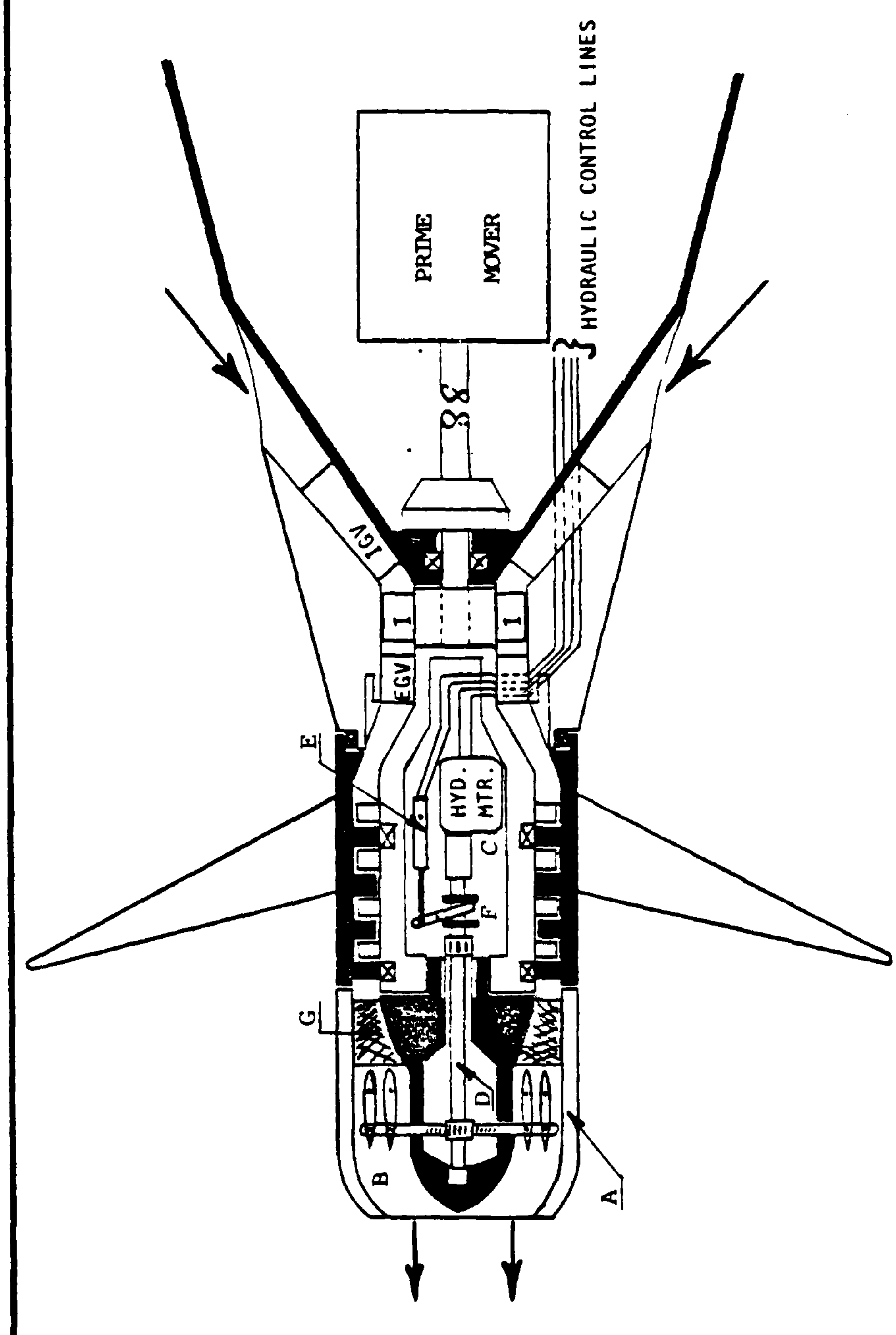


FIGURE 71. Ship Controlling Propulsion System

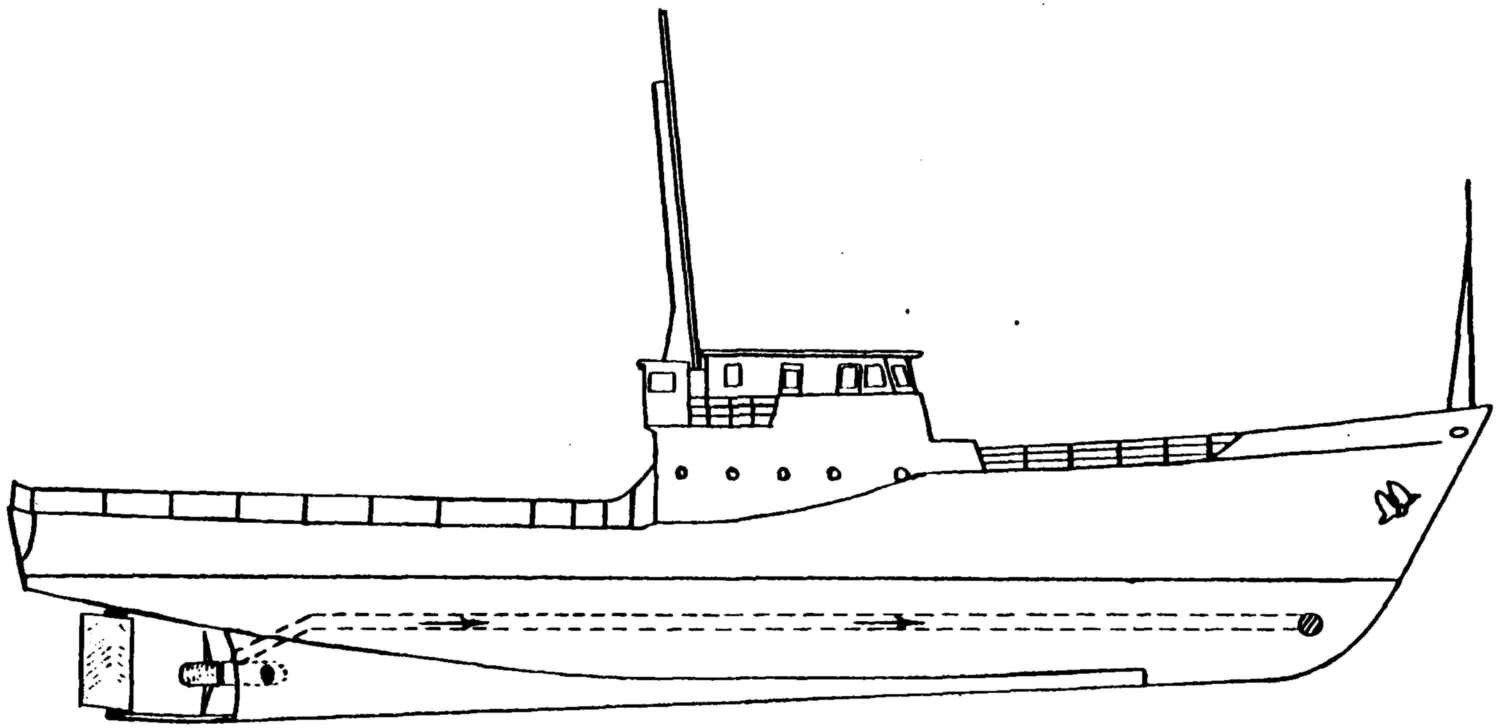


FIGURE 72. Installed Ship Controlling Propulsion System

TABLE 1.

Propeller Efficiency as a Function of Diameter Ratio

For B4-40

η_{max}	B_p	N_{rpm}	δ	D_{ft}	N/N_{70}	A	A_{70}/A	D/D_{70}
70%	12.6	427	145	9.5	1.000	70.882	1.000	1.000
72%	10.0	339	130	10.7	0.794	89.920	0.788	1.126
74%	7.8	264	114	12.1	0.618	114.990	0.616	1.274
76%	5.9	200	101	14.1	0.468	156.145	0.454	1.484
78%	4.0	135	84	17.4	0.316	237.787	0.298	1.832

For B3-35

η_{max}	B_p	N_{rpm}	δ	D_{ft}
79%	4.42	150	93	17.4
80%	3.50	119	82	19.3

Analysis of propeller performance was made using Wageningen B-series propellers and $B_p - \delta$ charts.

The variations in rpm, diameter, and propeller loading were investigated as they affect the propeller efficiency. Starting with V_A = 28 knots and P = 15,000 HP, the above values were derived.

TABLE 2.

CONTRAROTATING PROPELLER EFFICIENCY

Data From Comstock, p. 434

1. Efficiency increase -- 9%

2. Diameter decrease -- 15%

3. Weight increase -- 10% - 15%

TABLE 3.

PUMP 10 DETAILED SPECIFICATION

$A = \frac{12.5664}{3.8013}$ inch^2

$D_T = 4.0''$

$D_H = 2.2''$ $\quad A = 8.7651 = 0.060869$ Ft^2

$Q = 2.30$ ft^3/sec $= 1032.31$ gal/min $\quad C_m = 37.786 \quad \omega = 785.398$

$U = 130.9$ ft/sec $\quad U^2/2g = 266.07$

$H = 76.644$ Ft $\quad \Psi = H/(U^2/2g) = 0.288 \quad \varphi = 0.289$

$$N = \frac{2.30 \times 62.4 \times 76.644}{550} = 20.0 \text{ HP}$$

$$n_s = 125 \frac{1.5166}{13.517 \times 25.904} = 0.5414 \qquad \sigma = 1.14$$

$$H_{TH} = \frac{H}{0.85} = 90.17 \text{ Ft} \qquad N_s = 7500 \frac{32.129}{25.904} = 9302$$

$$\Delta C_u = \frac{90.17 \times 32.2}{U} \quad \frac{2903.5}{U} \qquad tg\,\beta_\infty = \frac{C_m}{U - \frac{\Delta C_u}{2}} \qquad tg\,\beta_1 = \frac{C_m}{U}$$

$$tg\,\beta_2 = \frac{C_m}{U - \Delta C_u}$$

$$GV \quad tg\,\alpha_1 = \frac{\Delta C_u}{C_m} \qquad W_\infty = \frac{C_m}{\sin\beta_\infty} \quad W_1 = \frac{C_m}{\sin\beta_1} \quad W_2 = \frac{C_m}{\sin\beta_2}$$

No.	Turbine	Z	γ	Q	H	D_R	D_T	n_R	C_m	U_T	ϕ	ϕ_T	ϕ_m	ϵ_m	U_m/C_1
1	T-1-1	8	0.6667	2.151	10.24	4.0	6.0	0.237	19.72	19.63	1.0046	1.71	2.464	48.31	-----
2	T-1-2	7	0.6667	2.151	11.707	4.0	6.0	0.214	19.72	19.63	1.0046	1.96	2.82	54.58	-----
3	T-1-3	5	0.6667	2.151	16.39	4.0	6.0	0.167	19.72	19.63	1.0046	2.73	3.94	72.91	-----
4	T-1-4	7	0.538	2.151	11.707	3.5	6.5	0.214	13.148	21.27	0.618	1.66	2.82	70.53	0.691
5	T-1-5	7	0.694	2.151	11.707	4.5	6.5	0.214	17.925	21.27	0.843	1.66	2.32	52.07	0.680
6	T-1-6	7	0.714	2.151	11.707	5.0	7.0	0.214	16.43	22.91	0.7172	1.436	1.96	48.27	0.772
7	T-1-7	9	0.6667	2.151	9.106	4.0	6.0	0.259	19.72	19.63	1.0046	1.522	2.19	43.85	0.624
8	T-1-8	10	0.6667	2.151	8.195	4.0	6.0	0.280	19.72	19.63	1.0046	1.370	1.985	39.50	0.639
9	T-1-9	9	0.586	2.151	9.106	3.7	6.3	0.259	15.17	20.62	0.736	1.38	2.19	51.43	0.715
10	T-2-1	8	0.680	2.365	9.31	4.25	6.25	0.2668	20.655	20.453	1.010	1.434	2.03	40.36	0.618
11	T-2-2	9	0.667	2.365	8.2778	4.0	6.0	0.2914	21.70	19.63	1.1053	1.383	1.99	36.62	0.604
12	T-2-3	7	0.692	2.365	10.643	4.5	6.5	0.2414	19.708	21.27	0.927	1.515	2.115	44.73	0.665
13	T-3-1	7	0.640	2.72	9.29	4.0	6.25	0.2867	21.62	20.45	1.057	1.431	2.13	40.10	0.605
14	T-4-1	7	0.6875	2.50	9.76	4.4	6.4	0.2648	21.72	20.94	1.013	1.412	2.013	39.96	0.639
15	T-4-2	7	0.6563	2.50	9.76	4.2	6.4	0.2648	19.654	20.94	0.938	1.433	2.090	43.19	0.655
16	T-4-3	7	0.625	2.50	10.07	4.0	6.4	0.2648	19.259	20.94	0.920	1.478	2.24	44.64	0.645
17	T-4-4	7	0.656	2.50	10.07	4.20	6.40	0.2586	20.608	20.94	0.984	1.478	2.156	43.05	0.633
18	T-4-5	7	0.656	2.50	9.063	4.20	6.40	0.280	20.608	20.94	0.984	1.331	1.940	38.94	0.648
19	T-5-1	7	0.656	2.30	9.834	4.20	6.40	0.252	18.96	20.94	0.905	1.447	2.11	44.66	0.6666
20	T-6-1	7	0.640	2.30	9.854	4.00	6.25	0.252	19.25	20.45	0.941	1.517	2.26	46.03	0.639
21	T-7-1	7	0.772	2.30	9.854	4.40	5.70	0.233	32.1	18.65	1.813	1.824	2.322	30.0	0.432
22	T-7-2	6	0.772	2.30	10.22	4.40	5.70	0.208	32.1	18.65	1.81	1.89	2.41	33.2	0.430
23	T-7-3	5	0.772	2.30	12.26	4.40	5.70	0.181	32.1	18.65	1.81	2.27	2.89	37.1	0.420
24	T-7-4	4	0.772	2.30	15.33	4.40	5.70	0.151	32.1	18.65	1.81	2.84	3.61	45.6	0.403

TABLE 4. Turbine Design Summary

TABLE 5. Blade Details, Turbine 23 and 24 Rotor Data

7-3 5-Stages								
D	U	C_m'	ΔC_u	C_{1u}	C_{2u}	$tg\beta_1$	β_1	$\sin\beta_1$
5.70	18.65	33.81	21.173	19.912	1.262	1.698	59.50	0.8617
5.05	16.53	33.81	23.888	20.209	3.679	1.673	59.13	0.8584
4.40	14.40	33.81	27.422	20.911	6.511	1.617	58.26	0.8505
D	$tg\beta_2$	β_2	$\sin\beta_2$	C_1	C_2	α_1	α_2	Σ
5.70	26.791	87.86	0.9993	39.24	33.83	30.50	2.14	32.64
5.05	9.910	83.79	0.9941	39.39	34.01	30.87	6.21	37.08
4.40	5.193	79.10	0.9820	39.75	34.43	31.74	10.90	42.64
7-4 4-Stages								
D	U	C_m'	ΔC_u	C_{1u}	C_{2u}	$tg\beta_1$	β_1	$\sin\beta_1$
5.70	18.65	33.81	26.466	22.558	3.908	1.499	56.29	0.8318
5.05	16.53	33.81	29.860	23.195	6.665	1.458	55.55	0.8246
4.40	14.40	33.81	34.277	24.399	9.939	1.386	54.18	0.8109
D	$tg\beta_2$	β_2	$\sin\beta_2$	C_1	C_2	α_1	α_2	Σ
5.70	8.651	83.41	0.9934	40.65	34.03	33.71	6.59	40.30
5.05	5.073	78.85	0.9811	41.00	34.46	34.45	11.15	45.60
4.40	3.402	73.62	0.9594	41.69	35.24	35.82	16.38	52.20

TABLE 6.

Cascade Blade Configuration

	Row Thickness b (inch)		Pitch τ (inch)			HUB		MEAN		TIP			
	Design	Actual	hub	mean	tip	Blade Outlet Angle δ_m (calc.)	Throat Θ (inch)	Blade Outlet Angle δ_m (calc.)	Throat Θ (inch)	Blade Outlet Angle δ_m (calc.)	Throat Θ (inch)	Ratio Θ	Hydraulic Diameter (ft.)
29 Blades 5 Stage Stator	.582	.540	.433	.555	.677	51.0°	.2720	49.7°	.3595	48.5°	.4500	.3544	.04514
29 Blades 6 Stage Stator	.581	.570	.433	.555	.677	45.8°	.3018	45.2°	.3912	42.7°	.4963	.3924	.04848
29 Blades 7 Stage Stator	.609	.590	.433	.555	.677	40.8°	.3278	39.9°	.4256	39.9°	.5192	.4202	.05105

tip dia. =6.25 in.   mean dia. =5.13 in.   hub dia. =4.00 in.

TABLE 7.

Cascade Data and Nozzle Efficiency

stator number	N_{re}	(RMS) inlet P_{tot}(in-H_2O)	(RMS) exit P_{tot}(in-H_2O)	nozzle efficiency (%)	exit velocity measured (ft/sec)	exit velocity using δ_s	error (%)	(RMS) δ_s (deg)
7	27586.1	2.753	2.3592	85.67	72.72	73.30	.79	26.89
7	35937.5	4.606	3.97	86.2	95.62	94.89	.76	26.9
7	49823.3	8.51	7.54	88.58	132.05	132.65	.45	29.56
6	28718.7	2.68	2.46	91.63	74.99	78.63	4.6	36.46
6	36365.5	4.50	4.12	91.75	97.85	100.77	2.9	36.69
6	42567.7	6.46	6.03	93.33	115.06	121.01	4.9	36.87
5	28908.2	2.90	2.73	94.35	81.14	84.99	4.5	45.06
5	36886.4	5.26	4.90	93.25	110.50	111.88	1.24	45.22
5	40250.5	6.58	6.17	93.67	124.99	125.48	.38	45.06

TABLE 8.

Air Turbine Characteristics at Design Point

AIR TURBINE CHARACTERISTICS AT DESIGN POINT

turbine	N_{re}	flowrate (CFS)	efficiency (total)
T5525	15081	6.52	.687
T5535	20643	9.13	.745
T5545	26491	11.73	.919
T5553	31156	13.82	.928
T6525	17556	7.17	.741
T6535	22807	10.04	.880
T6545	28806	12.90	.941
T7525	19918	7.79	.875
T7535	27214	10.91	.914
T7538	30051	11.85	.914

TABLE 9.

Pump 1 Maximum Efficiency Data

SPEED n RPM	FLOW COEFFICIENT ϕ	EFFICIENCY VALUE η %	EFFICIENCY TYPE
3000	0.317	94.4	STAGE
4500	0.304	87.2	STAGE
5000	0.307	85.7	STAGE
6000	0.315	87.0	STAGE
3000	0.303	90.9	TOTAL
4500	0.300	86.0	TOTAL
5000	0.284	84.0	TOTAL
6000	0.315	85.7	TOTAL
3000	0.271	82.2	SYSTEM
4500	0.268	74.2	SYSTEM
5000	0.280	77.0	SYSTEM
6000	0.315	81.2	SYSTEM

TABLE 10.

COMPARATIVE RESULTS OF PUMP 1 AND PUMP 2

		PUMP 1	PUMP 2	
	ϕ	.30	.24	- 20% Difference
	ψ	.27	.25	- 7% Difference
	Specific Speed	.578	.548	
MAXIMUM	3000 RPM	90.9	*	
TOTAL	4500 RPM	86.0	84.1	
EFFICIENCY	5000 RPM	84.0	82.0	
	6000 RPM	85.7	76.7	
MAXIMUM	3000 RPM	94.4	*	
STAGE	4500 RPM	87.2	84.8	
EFFICIENCY	5000 RPM	85.7	82.2	
	6000 RPM	87.0	77.0	
MAXIMUM	3000 RPM	82.2	*	
SYSTEM	4500 RPM	74.2	77.6	
EFFICIENCY	5000 RPM	77.0	77.6	
	6000 RPM	81.2	72.1	

* Not Available

TABLE 11.

Optimum Transmission Performance

5-Stage Turbine
Long Exit Guide Vanes
Bullet Diffuser
Pump 1

	PUMP SPEED					
	3000	3500	4000	4000*	4500	4500*
Transmission						
Transmission Ratio, TR	5.9	5.9	6.0	6.0	6.0	6.0
Transmission Efficiency, η_{TR}	65	65	63	65	57	62
Pump						
Flow Coefficient, ϕ_P	.215	.216	.216	.214	.217	.218
Head Coefficient, Ψ_P	.370	.371	.370	.370	.371	.371
Input Power (Hp)	1.67	2.74	4.31	4.20	6.75	6.32
Efficiency, η_P (%)	77	74	70	71	64	69
Specific Speed, n_s	.387	.387	.388	.386	.388	.388
Cavitation Number, σ_{BL}	1.29	.95	.73	1.09	.59	.86
Turbine						
Speed (RPM)	508	593	667	667	750	750
Flow Coefficient, ϕ_T	.759	.767	.774	.764	.778	.779
Stage Head Coefficient, Ψ_{STG}	1.20	1.24	1.25	1.23	1.25	1.23
Output Power (Hp)	.97	1.58	2.44	2.47	3.50	3.53
Efficiency, η_T (%)	83	86	89	90	90	89
Reynolds Number (x 10^5)	3.9	4.5	5.2	5.2	5.9	5.9

*Pressurized

TABLE 12.

Optimum Transmission Performance

5-Stage Turbine
Long Exit Guide Vanes
Improved Diffuser
Pump 1

	PUMP SPEED				
	3000	3500	4000	4500	5000*
Transmission					
Transmission Ratio, TR	6.5	6.5	6.5	6.5	6.5
Transmission Efficiency, η_{TR}	67	63	59	56	60
Pump					
Flow Coefficient, ϕ_P	.210	.212	.212	.209	.210
Head Coefficient, Ψ_P	.370	.369	.370	.370	.370
Input Power (Hp)	1.63	2.72	4.31	6.44	8.40
Efficiency, η_P (%)	77	74	68	64	67
Specific Speed, n_s	.383	.385	.384	.381	.382
Cavitation Number, σ_{BL}	1.30	.95	.73	.58	.65
Turbine					
Speed (RPM)	461	538	615	692	769
Flow Coefficient, ϕ_T	.816	.826	.823	.812	.820
Stage Head Coefficient, Ψ_{STG}	1.460	1.473	1.463	1.471	1.469
Output Power (Hp)	.97	1.53	2.35	3.26	4.67
Efficiency, η_T (%)	86	84	87	86	88
Reynolds Number (x 10^5)	3.8	4.4	5.1	5.6	6.3

*Pressurized

TABLE 13.

Optimum Transmission Performance

5-Stage Turbine
Long Exit Guide Vanes
Bullet Diffuser
Pump 2

	PUMP SPEED				
	3000	3500	4000	4500	5000*
Transmission					
Transmission Ratio, TR	7.2	7.2	7.2	7.2	7.2
Transmission Efficiency, η_{TR}	69	68	67	64	63
Pump					
Flow Coefficient, ϕ_P	.190	.190	.190	.191	.192
Head Coefficient, Ψ_P	.327	.328	.328	.327	.328
Input Power (Hp)	1.10	1.94	3.00	4.63	6.25
Efficiency, η_P (%)	93	83	80	74	75
Specific Speed, n_s	.398	.398	.398	.400	.400
Cavitation Number, σ_{BL}	1.31	.97	.74	.58	.71
Turbine					
Speed (RPM)	417	484	555	625	694
Flow Coefficient, ϕ_T	.820	.819	.821	.824	.821
Stage Head Coefficient, Ψ_{STG}	1.68	1.64	1.65	1.71	1.65
Output Power (Hp)	.68	1.20	1.80	2.63	3.54
Efficiency, η_T (%)	73	81	83	85	83
Reynolds Number (x 10^5)	3.4	4.0	4.6	5.2	5.9

*Pressurized

TABLE 14.

Transmission Performance Comparing Pump #1 and Pump #2

	Pump #1	Pump #2
Pump Φ at Peak Transmission Efficiency	.215	.192
Optimum Transmission Ratio	6:1	7:1

Peak Transmission Efficiency

Pump RPM	Pump #1	Pump #2
3000	65	71
3500	65	67
4000	63	67
4500	62	64.5
5000	*	63.3
6000	*	*

* not available

TABLE 15.

REVERSE THRUST MECHANISM PERFORMANCE

Pump Speed (rpm)	1000	2000	3000	4000
Flow (ft^3/sec)				
Forward*	.225	.464	.730	.990
Reverse	.149	.314	.468	.634
Pump Flow Coefficient				
Forward*	.210	.218	.228	.237
Reverse	.138	.146	.158	.148
Reversing Thrust (lbf)	1.61	7.16	15.9	29.2
Reversing Power (Hp)	.010	.096	.317	.790
Turbine Power (Hp)	.036	.287	.970	2.44
R (%)	40	48	47	46

*No load on turbine

TABLE 16.

PRELIMINARY WEIGHT ESTIMATE FOR REGULAR PROPELLER AND NEW PROPULSION SYSTEM

A WEIGHT FOR CONSTANT PROPELLER DIAMETER

	REGULAR PROPELLER		NEW PROPULSION SYSTEM
1 PROPELLER WEIGHT	W_P	$=$	W_P'
	SOLID HUB	→	HOLLOW HUB INCLUDES TURBINE BLADE AND SUPPORT SHAFT
2 WEIGHT OF SHAFT, SEAL, BEARINGS			
	SHAFT - LARGE DIAMETER LONG	>	SHAFT SMALL DIAMETER SHORT
	SEAL - LARGE	>	SEAL - SMALL
	BEARINGS - LARGE	>	BEARINGS - SMALL
			ADD: PUMP - HOUSING - GUIDE VANES TRANSITION TO TURBINE - ALL HOLLOW AND IN WATER

$$W_{SSB} \approx W'_{SSB}$$

B WEIGHT SAVINGS:

GEAR
FOUNDATION FOR GEAR
REVERSING MECHANISM

ABOUT:
<u>1.5 GEAR WEIGHT</u>

TABLE 17.

Potential Improvements for the New Propulsion System Due to Weight Savings

C POTENTIAL IMPROVEMENTS

USE SOME OF THE WEIGHT SAVINGS FOR:

1 CONTRAROTATING PROPELLER

2 LOWER RPM FOR HIGHER EFFICIENCY

TABLE 18.

SPECIFIC GEAR COSTS

Horsepower	\$/lb Commercial	\$/lb Military
10,000	15	20
20,000	13.4	17.6
30,000	11.7	14.8
40,000	10	12
50,000	10	12

TABLE 19.

GEAR COSTS AS A FUNCTION OF SHAFT SPEED

Horsepower	$ x 1000 C-80		$ x 1000 C-100		$ x 1000 C-120		$ x 1000 C-150
10,000	1980	----	1620	----	1260	----	1005
20,000	2879	2266	2333	1866	2000	----	1514
30,000	----	2707	2800	2240	2381	1937	1907
40,000	----	----	----	----	2500	2060	----

Horsepower	$ x 1000 M-80		$ x 1000 M-100		$ x 1000 M-120		$ x 1000 M-150
10,000	2640	----	2160	----	1680	----	1260
20,000	3743	2946	3033	2426	2600	----	1988
30,000	----	3403	3521	2817	2993	2435	2412
40,000	----	----	----	----	3000	2472	2520

C = Commercial
M = Military

Data calculated from Figures 10 and 11.

TABLE 20.

HYDRAULIC TRANSMISSION COSTS

Gas Turbine GE-LM 2500	25,000 HP	$1,837,000
60%		$1,100,000
GE-LM 1500	15,000 HP	$1,200,000
60%		$ 725,000

Hydraulic transmission does not have:

Combustion chamber, nozzles, fuel control, hot section
Fuel treatment system
2 flexible joints and long power transmission shaft
Exhaust ducts
Compensation for axial and radial expansion due to heat expansion
Therefore, assume cost equals 60% of gas turbine cost.

TABLE 21.

TRANSMISSION PERFORMANCE WITH INCREASED SPEED

Number	Prime Mover and Pump RPM	Propeller RPM	Power HP	Speed Factor	Power Factor
1	7,500	750	15	1	1
2	10,000	1,000	35	1.33	2.36
3	15,000	1,500	120	2	8
4	20,000	2,000	285	2.67	19
5	22,500	2,250	405	3	27
6	25,000	2,500	555	3.33	37
7	30,000	3,000	960	4	64